ENGINEERING TOLERANCES

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A STUDY OF TOLERANCES, LIMITS AND FITS FOR ENGINEERING PURPOSES, WITH FULL TABLES OF ALL RECOGNIZED AND PUBLISHED TOLERANCE SYSTEMS

BY

H. G. CONWAY

M.A., A.M.I.MECH.E., F.R.AE.S.

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PREFACE

THE use of tolerances, limits, and fits in engineering design is an essential factor in modern manufacture, whether by mass, batch, or even job production. It is surprising, therefore, that there appears to be no publication which explains the scope of the various published limit systems, and endeavours to instruct the designer in the proper use of such systems.

The gulf between designer and shop man—a gulf into which many platitudes have been cast—is not made less by the ignorance of the average designer of the meaning and significance of tolerances and fits, nor of the practical man of interchangeability requirements.

This book, therefore, is an attempt at a clarification of the position, the prime object being to persuade the designer to use tolerances and limits with sincerity and genuine intention, both shop man and inspector then being in a position to treat the designer's drawings with more respect than is current in most production shops to-day.

The author feels, as a result of some study and experience, that modern inspection methods have not kept pace with general technical and production development. The accuracy of absolute measurement which is possible to-day is, of course, very high indeed, but the inspection methods used in the average shop leave much to be desired from the point of view of accuracy in the hands of semi-skilled personnel. It is probable that few designers realize the extent of the errors introduced by the comparatively crude systems of measurement (plug and gap gauges, etc.) which are being used in their production shops. Since it is the final result and its performance in the field which count, it is not suggested that these errors are necessarily serious, but a proper understanding between designer and producer can only come when the truth is known.

In the author's experience a serious defect in the training of young designers is that they have no ready means of knowing or finding out what are the practical limitations of accuracy of the machines which are to be used to make the components they are designing. Some attempt has, therefore, been made at stating, with what authority the author's experience may entitle him to, on the one hand, and a considerable research into published material where and if available, on the other hand, the inherent limitations of the main types of machine and production processes. It is known, for example, that a skilled toolroom milling machine operator can, and does, mill to

"half a thou.," probably less. This has no relevance, however, because the designer should not be concerned with what *can* be done, but with what is reasonable to expect from his production shop, making full allowance for wear of tools, a lower grade of skilled labour, the condition of the plant, etc. For a designer to expect a 4 in. lug to be milled in width to less than, say, 0·002 in. is, therefore, entirely wrong. Where tolerance qualities or grades are quoted in this book for various processes, they are certainly not the ultimate in possibility, but are quite definitely put forward as the limit to which the designer should go.

The tolerance tables which are as comprehensive as possible and which will, it is hoped, be found of value for convenience of reference, have been checked with care several times to eliminate errors; it is a pity that the restrictions of the present day have made it impossible to set some of the more complicated tables with horizontal ruling to assist the eye in following a line of figures. As explained in the text, however, a designer should extract from the complete tables those fits which most apply to his particular products and will, therefore, prepare his own abbreviated tables.

Acknowledgments are gratefully paid to the British Standards Instution, 24 Victoria Street, London, S.W.1, for permission to reproduce from some of its standards; to the Newall Company for assistance with information on its limit system and for the supply of illustrations; to Messrs. Chesterman, J. E. Baty, Ltd., Alfred Herbert, Solex and General Motors for similar assistance with other illustrations. Other sources of information, particularly on special processes, are indicated at the appropriate place. If the reader is unable to find information about a particular process it is probable that this is because information is simply not yet available. As far as possible, such omissions will be rectified in a subsequent edition.

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CHAPTER I

FUNDAMENTALS

FUNDAMENTALS OF THEORY

1. Definitions

THE very first essential for the correct use and comprehension of engineering tolerances is to understand the correct meanings of the various terms in common use. The definitions that follow give the accepted sense of the principal terms in general use without pretending to be particularly brief. (See Chapter III (Quality Control) for other definitions.)

Nominal Size. The size of a component, etc., to the nearest standard or round figure without regard to any special limits—e.g. 0·374/0·373 in. is nominally $\frac{3}{8}$ in.

Limits. The extreme values of size of a dimension. Thus—

Upper limit, the greatest size.
Lower limit, the smallest size.

Tolerance. The tolerance or latitude given to the production shop in achieving a given dimension, i.e. the difference between upper and lower limit. *Grade* or *quality* of tolerance refers to one of a range of tolerances of varying severity or magnitude.

Unilateral Limits. A tolerance expressed with one limit (generally the lower limit of a hole) as zero, i.e. the nominal size.

Bilateral Limits. A tolerance expressed with one limit positive and the other negative (e.g. Newall A or B holes).

Deviation or Allowance. The difference of size of male and female components, or in other words the physical dimension determining a particular fit. The *basic deviation* is the most essential difference in size which determines a fit, i.e. the minimum clearance or interference, and is the factor most important to the designer.

Fit. A fit is the practical result of mating two parts (e.g. a shaft in a hole). Fits are usually classified broadly as interference or press fits, transition fits and clearance or working fits.

Press fits occur when the male member is larger than the female.

Transition fits occur when the resulting fit, due to variation in size of male and female components owing to their tolerances, varies between clearance and interference.

Clearance fits occur when the male member is smaller than, and thus loose in, the female member.

Fig. 1 illustrates diagrammatically these definitions as applied to a unilateral hole and a clearance shaft. Fig. 2 shows the tolerance

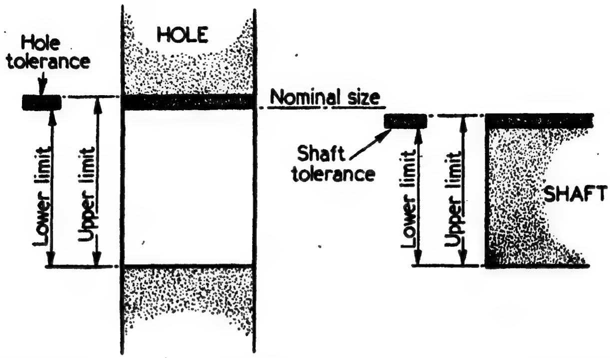


FIG. 1. DIAGRAMMATIC REPRESENTATION OF LIMITS AND TOLERANCES UNILATERAL HOLE AND CLEARANCE SHAFT

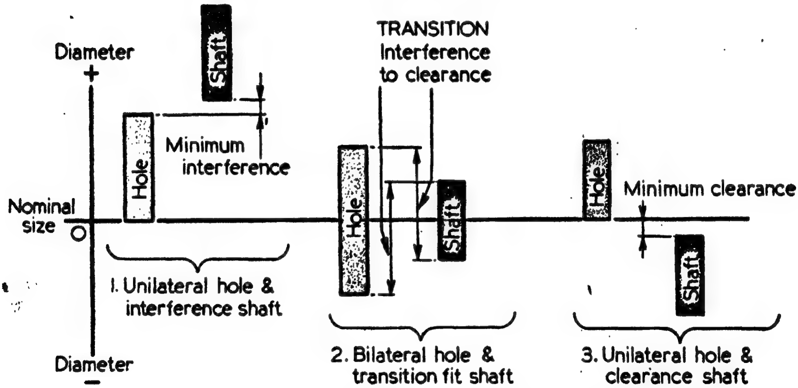


FIG. 2. DIAGRAMMATIC EXAMPLES OF FUNDAMENTAL TYPES OF DIFFERENT FITS

zones of various types of fit in enlarged detail. The use of diagrams of the latter type when considering tolerance problems is of great value in the rapid comprehension or visualization of a fit. In later chapters diagrams of this type are used to illustrate the various types of fit.

2. Fundamental Considerations

If components could be manufactured to exact sizes, questions of *tolerance* would not arise, and the *limits* would merge into a

single *deviation* of size from the nominal. Thus, a press fit might have a shaft deviation of + 0·001 in. as compared with the hole, giving 0·001 in. interference. It will be seen therefore that the fundamentally important limits are—

On clearance fits, those determining the minimum clearance. ✓

On interference fits, those determining the minimum interference.

In the case of a clearance fit, the manufacturing tolerance adds to the clearance and thus determines the amount of departure from the minimum. In the case of press fits the tolerance increases the severity of the interference. Thus the class of fit is considered to be

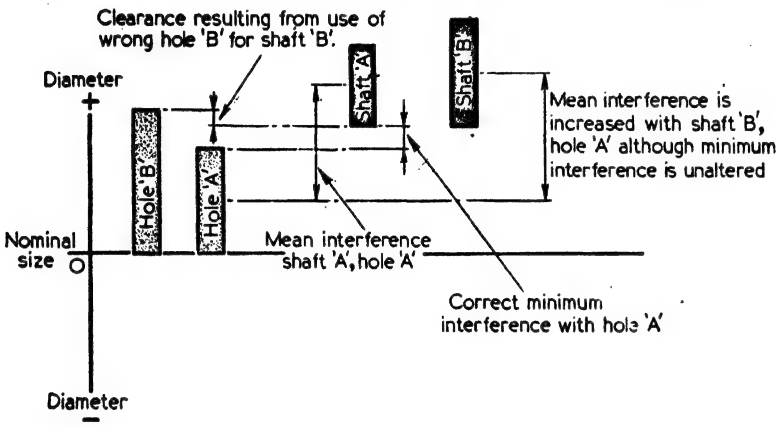


FIG. 3. THE CORRECT ASSOCIATION OF HOLE AND SHAFT IS IMPORTANT WITH PRESS FITS

independent of the manufacturing tolerance; although from the designer's point of view this is correct, it is true, however, that the tolerance affects the average or mean size and thus the average performance of the product, if not the one extreme case. (This aspect is considered in detail later in Chapter III.)

Fig. 3 illustrates that a press fit shaft with a normal unilateral (or bilateral) hole is correct only with a particular maximum grade of shaft (although one with a smaller tolerance would be satisfactory).

Fig. 4 illustrates that with clearance fits and unilateral holes, increase of tolerance on either hole or shaft does not affect the minimum clearance and thus alternative grades of either may be mated if desired. This in practice is very useful but is achieved in the full only by the I.S.A. Tolerance System. It is very convenient, when a certain combination of hole and shaft is proving too severe in production, to alter the grade of hole or shaft, or both, with the knowledge that the class of fit is unaltered.

As regards transition fits, these by definition are neither one type of true fit nor the other, the exact fit depending on the actual size of the parts, and alteration in tolerance affects the variation. In practice several fits may exist in more than one grade, for combination with different grades of the other member, but the variation of fit may be such that itsclassification or description may be impossible.

Excluding "lapped" fits, the smallest tolerances are found on

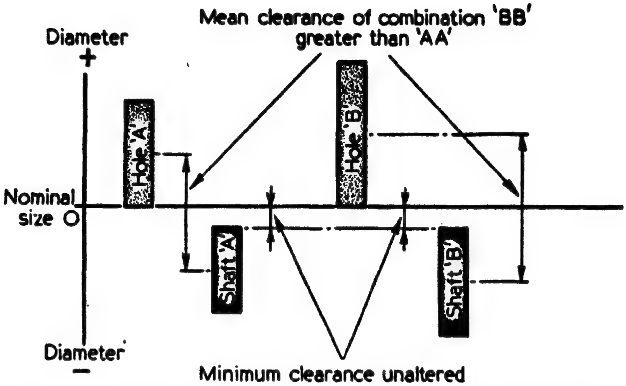


Fig. 4. The Effect of Tolerance on a Clearance Fit

press fits, the greatest on clearance fits, with the intermediate transition fits between the two.

The designer must therefore in determining limits for a product proceed as follows—

(i) Settle the class of fit required by considering the minimum clearance or interference required, taking into account the design, questions of lubrication, etc.

(ii) Settle the manufacturing tolerance to be allowed to the factory, with very particular reference to available facilities.

(iii) Finally consider the resultant variations of fit and determine if the mean fit is satisfactory from a functioning and service life point of view.

In subsequent chapters it is pointed out that determination of limits is a highly skilled job and a strong recommendation is made that comprehensive selection tables be prepared for use in the Drawing Office, leaving the determination of fits to the more qualified and experienced engineer. The usual result of unskilled selection is excessive severity.

3. Meaning of Limits

Limits are often put on drawings without regard to what is really meant by them. If a drawing specifies a shaft as 1·0 + 0·002 + 0 in., usually what is meant is that the shaft may have a diameter of between 1·000 and 1·002 in., but do these dimensions refer to the

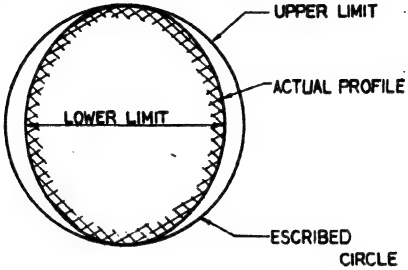


FIG. 5. THE MEANING OF LIMITS ON AN IMPERFECT DIAMETER

mean diameter if it is oval or tapered, and, if so, what tolerance is there on the ovality or taper?

Fig. 5 shows a section of a shaft of imperfect shape. The correct interpretation of the limits on this diameter, in the absence of more specific information, is that the upper limit measures the maximum diameter of the shaft *at any point* and the lower limit the smallest

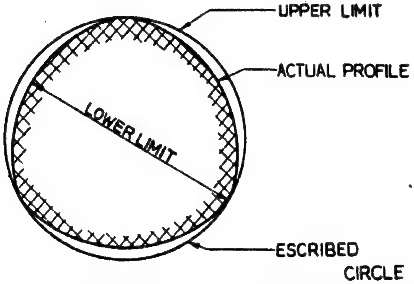


FIG. 6. THE "LOBING" OR TRIANGULATION TYPE OF ERROR

diameter at any point, the word diameter being used in a loose sense, as the actual profile may not be circular.

It is quite common, especially with centreless grinding, to produce cylindrical parts whose actual profile is the lobe-shaped triangular figure shown in Fig. 6. It is quite possible to generate a shape of constant breadth which would appear to be within drawing tolerance when measured with an ordinary gap gauge, and yet the part might depart widely from the designer's intention,

since it would not enter a hole whose diameter was equal to or even slightly larger than the upper limit.

By similar reasoning, it can be seen that in the case of limits on a hole, more is meant than simply the maximum and minimum sizes of the hole. The errors of the hole might well be as shown (exaggerated) in Fig. 7, and yet a simple measurement of diameter would not indicate the lack of axial truth.

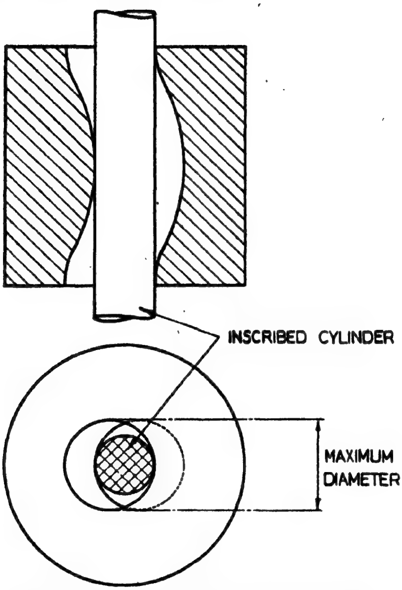


FIG. 7. AXIAL ERRORS ON A HOLE

The correct interpretation of a pair of limits can thus be stated as follows—

Hole: the upper limit refers to the greatest diameter at any point in the hole; the lower limit refers to the diameter of the inscribed cylinder which will just pass through the hole.

Shaft: the upper limit refers to the diameter of the ex-scribed cylinder which will just pass over the shaft; the lower limit refers to the minimum diameter at any point on the shaft.

When applying limits to non-circular parts, with flats, slots, key-ways, etc., a similar interpretation should be made to cover errors of truth or profile in an axial direction.

Although, strictly speaking, this interpretation should apply to holes in long components ($L/D > 3$), it is unreasonable to expect such axial accuracy in, say, a cylinder bore 12 in. long and 2 in.

in diameter, and generally the design can cater for small amounts of error. The normal tolerance could apply to the greatest diameter and the inscribed *circle* (not cylinder) or at least to an inscribed cylinder of length less than the bore in question. This matter, however, is one to be solved on the merits of the case.

In applying selective assembly to modern production, it often occurs that the tolerance on the diameter of a part is greater than the permissible departure from profile accuracy, because the part will, after manufacture, be graded for selective assembly. Thus, a piston pin in a motor car piston may be ground to a tolerance of 0·001 in., but the surface accuracy may have to be held to the equivalent of a diameter tolerance of 0·0005, since the pins are graded before assembling in a piston, and since the resultant fit is precise and important. In this case the necessary information will have to be specially stated on drawings.

The fact that two or more diameters are each finely toleranced does not imply that the concentricity of each is limited unless specifically stated. Concentricity tolerances are usually specified only on those dimensions which are important or where the normal manufacturing process is known to lead to errors which must be checked. In general, too little attention is paid by designers, in the application of limits to drawings, to concentricity and truth—questions affecting the final result required, as for example with ball race fits.

Although normally limits refer to the extreme dimensions which may ever occur on a component, a more recent outlook on the matter is outlined later in Chapter III, where the limits refer to the dimensions inside which all but a small percentage of the parts must lie. Provided this probability effect is taken into account, however, the definitions given above still hold good.

4. Tolerance Units

Tolerances are expressed in suitable fractions of the inch or millimetre.

The inch is generally divided into—

Thousandths	0·001
Ten-thousandths	0·0001
Twenty-thousandths . . .	0·00005

As the half-tenth (0·00005 = 50 micro-inches) is the smallest subdivision of standard slip blocks usually used in the setting of workshop gauges, limits are seldom expressed to greater accuracy. In fact, except for fine tolerances on small components, the usual unit is the tenth (0·0001).

The millimetre is invariably divided into microns (= 0·001 mm), symbol μ. For all normal engineering purposes a micron can be taken as 0·00004 in. ($\frac{4}{10}$ of a "tenth"), 25 microns (actually 25·4) being 0·001 in. The micron is the smallest unit on metric slip blocks and seldom if ever divided, except on work of extreme precision. Continental practice is not to work in rounded off figures greater

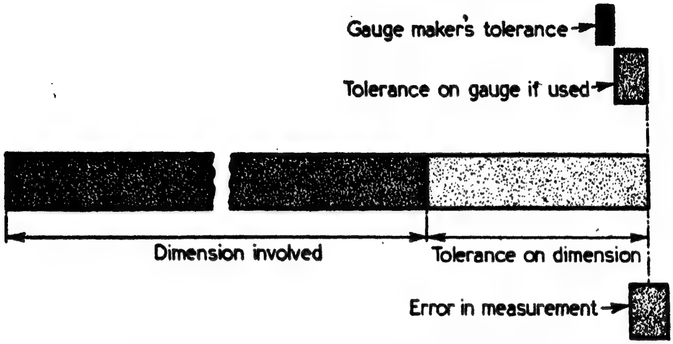


FIG. 8. WORK AND GAUGE TOLERANCES

than a micron, although the British Standard metric tolerances are often rounded off to the nearest 5 microns (0·005 mm).

GAUGING AND MEASUREMENT

1. Introduction

Whatever dimension an engineer specifies on a drawing, whether it is a round figure of 1 in. or a dimension with a limit reading 1·06255 in., it must be measured many times during the manufacturing process and often afterwards at the customer's works. It is obviously of prime importance for the production or inspecting engineer to use measuring equipment suitable for measuring with the necessary degree of accuracy to achieve the corresponding degree of dimensional accuracy. For example, in measuring the overall length of a long steel cylinder, a tape measure reading to $\frac{1}{16}$ in. would probably suffice, but at the other extreme a slip gauge may have to be measured to a few micro-inches.

A few moments' consideration will show that the accuracy of measurement (the production man's concern) must relate to the tolerance involved, which in turn will relate (the designer's concern) to the magnitude of the dimension being measured.

Fig. 8 will make this more clear. A dimension such as the length of a bar has a tolerance as shown diagrammatically. If this length is measured with a rule, caliper, micrometer, etc., there will be an

error in reading, depending in magnitude on the measuring method used. If a fixed gauge is used, the gauge itself must have a wear allowance. Finally, the manufacturer of the gauge itself must have a tolerance to which to make the new gauge.

A normal component tolerance is 0·001 per inch or 1 : 1000; the gauge setting allowance is usually 10 per cent of this. The gauge blockmaker's tolerance is usually 10 per cent of the result, or $0{\cdot}1 \times 0{\cdot}1 \times 0{\cdot}001 = 1 : 100{,}000$.

This matter is referred to in detail in Chapter VIII, as it is quite a complicated matter requiring special study. In modern gauging systems the actual limits allowed to pass through the gauge may fall slightly outside the drawing limits.

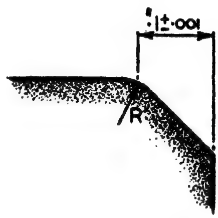


FIG. 9. A DIMENSION THAT CANNOT BE MEASURED

In subsequent paragraphs the methods and accuracy of measurement of various processes and instruments is considered, and Table 1 sets these out in convenient form. It is of great importance for the designing engineer to understand these problems, as drawings are too often produced with dimensions and limits specified which either cannot be measured with any normal method or else cannot be measured with the particular equipment available in the engineer's factory. Fig. 9 illustrates a dimension put on a drawing which cannot reasonably be measured.

TABLE 1

NORMAL ACCURACY OF MEASUREMENT BY VARIOUS METHODS

Method	Reading Error		Instrument Error		Total Error	
	in.	mm	in.	mm	in.	mm
Rule—12 in.	± 0·004	± 0·1	± 0·001	± 25μ	± 0·005	± 0·125
Vernier caliper—6 in.	± 0·001	± 25μ	± 0·0005	± 15μ	± 0·0015	± 0·04
Micrometer—0–6 in.	± 0·0002	± 5μ	± 0·0001	± 2μ	± 0·0003	± 7μ
Dial gauge—0·0001 in.	± 0·00005	± 1μ	± 0·0001	± 2μ	± 0·00015	± 3μ
Simple dial comparator	± 0·000025	± ½μ	± 0·00005	± 1μ	± 0·000075	± 1·5μ
Precision comparator	± 0·00002	± ½μ	± 0·00001	± 0·2μ	± 0·00003	± 0·7μ

1μ = 0·001 mm = 1 micron

One of the most difficult of all gauging problems is in the measurement of angles which invariably have to be reduced to linear dimensions; some components such as gears, splines, etc., cannot usually be measured in the normal sense of the word and are habitually inspected to agreed form gauges, although optical projection is used on many suitable parts such as threads, and on small instrument, clock, or watch components.

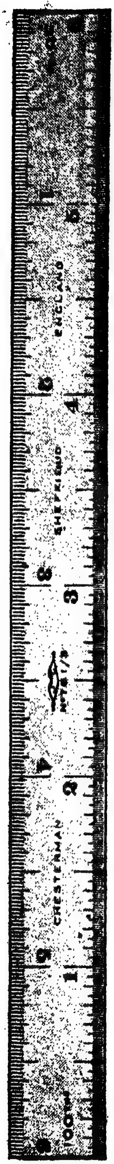


Fig. 10. The Rule

2. Elementary Methods

(a) The Rule

The rule (Fig. 10) is still the standard method of measuring lengths; an accurate rule can be read to the

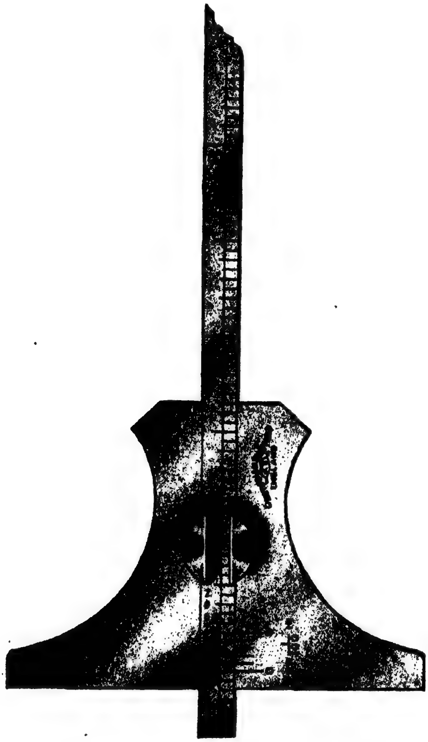


Fig. 11. The Depth Gauge

nearest 0·005 in. A depth gauge (Fig. 11) is merely a rule on a convenient mounting. Internal and external calipers can be used in conjunction with a rule to measure holes and shafts, etc., with a similar accuracy, but these tools are seldom used nowadays except for approximate measurement, or work on very large parts.

(b) Vernier Calipers

A vernier caliper (Fig. 12) can measure external diameters of all sizes, and holes larger than 0·25 in. (generally). By using the

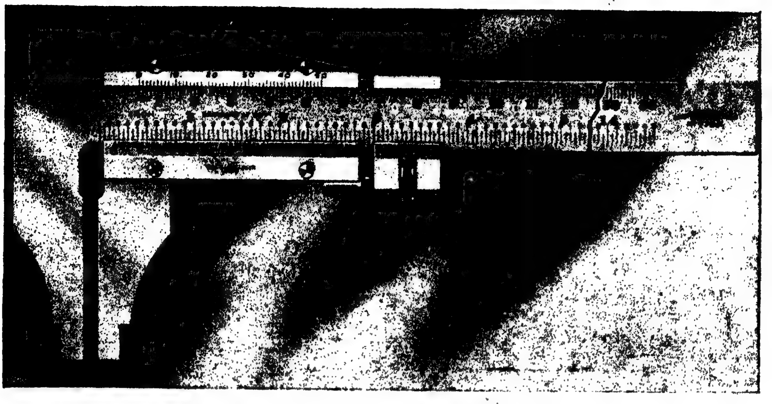


Fig. 12. The Vernier Caliper

vernier, usually examined with a magnifying glass, external diameters can be measured to 0·001 in. and internal diameters not quite so accurately (particularly as only the edge of the hole can be gauged). Vernier depth gauges and height gauges (Fig. 13) are particularly useful, the latter being the standard inspection method of measuring long dimensions.

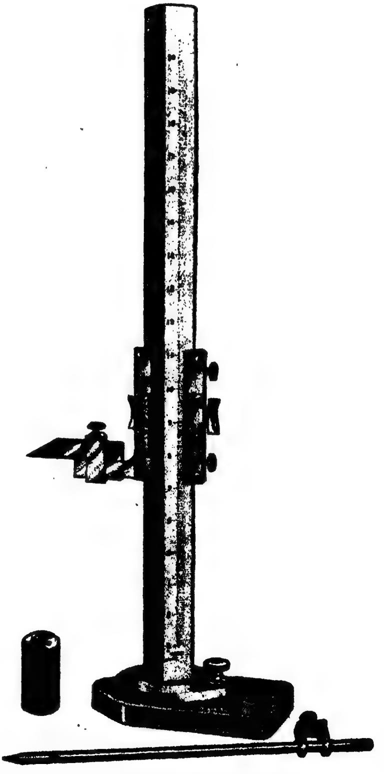


Fig. 13. The Vernier Height Gauge

3. Normal Shop Methods

(*a*) *Micrometers*

The micrometer is too well known to require much description. External micrometers are generally available in most engineering shops up to about 12 in., but micrometers for measuring a long length, say 24 in., are rather special, and often unsatisfactory owing to difficulties of handling, frame distortion, etc. Internal micrometers have a minimum limit of about $\frac{3}{4}$ in., but by means of the usual extension rods can measure

large bores quite satisfactorily. They are not used extensively as they are slow and delicate.

The normal inch micrometer is calibrated in 0·001 in. divisions, some having a $\frac{1}{10}$th vernier, reading to 0·0001 in. The normal metric micrometer has divisions of 10 microns or 0·0004 in., and is in fact easier to read accurately than the inch type. Although in a skilled inspector's hands a micrometer can read to a fifth

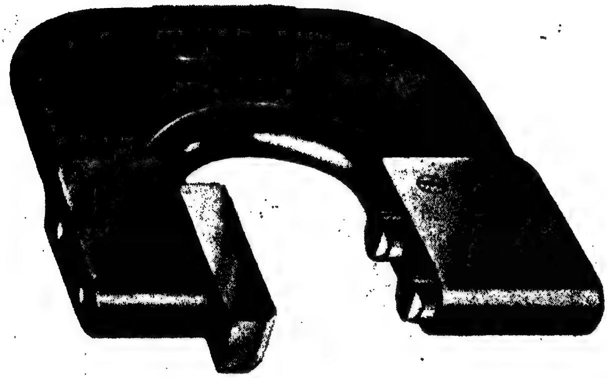


FIG. 14. THE SNAP OR GAP CALIPER GAUGE

(5 microns) this is exceptional, and to be sure of a half-thousandth is as much as can be relied upon.*

Some micrometers have a small friction screw device to regulate the screwing-up force in an endeavour to eliminate this particular variable, but it is seldom made use of since the main difficulty in using a micrometer is holding it truely square with the surface being measured, and the only satisfactory way is to pass it over the piece several times until the right feel is obtained.

Strictly speaking, a micrometer should be used with a specified load to make it pass the piece. In actual practice the skilled feel of the operator is relied upon.

(*b*) *Snap or Gap Gauges*

A normal snap or gap caliper gauge of the "Go" and "Not-Go" variety (Fig. 14) is a fixed micrometer set to some standard. It is not strictly speaking a measuring instrument but merely classifies according to its setting. It is an integral part of modern gauging

* See Appendix I.

methods, however, and with the plug gauge is the most common shop inspection device in a modern production shop.

The gap gauge measures the least diameter of the shaft under inspection and unless used with a ring gauge (see next section) does not really measure the drawing limits, as certain types of error, particularly triangulation, are missed. This is usually got over by an additional check with the shaft rotated in a vee-block under a dial gauge.

In an attempt to improve on the basis of measurement, some of these gauges have one anvil flat and of appreciable width, with the "Not-Go" anvil opposite also of appreciable length (a section of

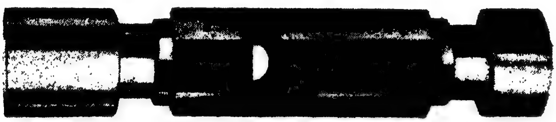


FIG. 15. THE PLUG GAUGE

a cylinder) so that the "Not-Go" check is along a line; the "Go" anvil is a single button or sometimes two buttons close together, so that the "Go" check measures a diameter. This system is sound if the flat anvil is of a length comparable with the part being measured.

Insufficient attention is invariably paid to the force required to pass a gap gauge. With normal handling, owing to the wedge action of the anvils passing over the part, a distortion of 0·0001 in. (2–3 microns) is quite easily obtained. For accurate measurement a gap caliper gauge must be used with a certain (light) maximum force, and the best practice is to use a special weighted holding balance. This, however, spoils the main advantage of a caliper gauge—that is, its portability—and where possible with accurate work, dial caliper gauges with built-in force limitation should be used.

(c) *Plug and Ring Gauges*

The standard method of measuring holes is the plug gauge (Fig. 15). The long end represents the smaller of the two limits and is thus the "Go" gauge. The short end (short because it rarely enters the hole) is the larger of the two limits and is thus the "Not-Go" gauge. Strictly speaking, the plug gauge does not measure the hole but merely determines whether it is between the specified

limits. When the hole size approaches its limit values, the ease of entry of the gauge depends on the force used and thus will give results depending on the method of handling. In practice it is quite reliable and is certainly widely used because of its low cost and ease of handling.

The plain cylindrical plug gauge is, however, open to the serious criticism that the "Not-Go" plug measures the inscribed circle and not the maximum diameter, which is what is required by the drawing tolerance. Correct practice would be to use a plug "Go" gauge and a pin "Not-Go" gauge. An equivalent method is to use a ball gauge on a handle (Fig. 16), inserted in the hole with the handle inclined, to act as a "Go" gauge. By moving the handle until it lies on the axis of the hole, a small pin projecting on the ball at right angles to the handle axis will thus read the maximum diameter or "Not-Go" size. In either case, the pin or ball gauge gauges the "Not-Go" limit if the handle of the gauge cannot be swung over from one side to the other, several attempts being made around the hole in an endeavour to find any ovality.

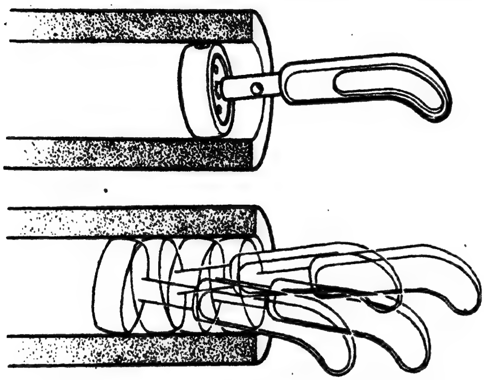


FIG. 16. THE "TE-BO" SPHERICAL BALL GAUGE

Although cylindrical "Not-Go" gauges are satisfactory when the machining process automatically takes care of ovality, many products have suffered from undetected ovality, and greater use of the pin type of gauge will probably be evident in the future.

The female equivalent of the plug gauge is the ring gauge, which is not used often, as a fixed caliper gauge is to be preferred.

A "Go" ring gauge does measure the ex-scribed circle of the shaft being measured and is thus a sound method of measuring if a "Not-Go" caliper gauge is used as well to measure the least diameter. The theoretical advantage of such a ring gauge is outweighed by its practical snags, difficulty of applying to the work, difficulty of manufacture and lack of adjustment for wear.

(*d*) *Dial Instruments*

There is a family of measuring instruments relying on the dial gauge—familiarly known as the "clock" (Fig. 17). These read the

movement of a plunger in terms of the rotation of a pointer around a dial calibrated in various degrees of accuracy down to 0·0001 in. or $2\frac{1}{2}$ microns, and can thus be read to less than half a scale division. However, the conditions of handling are rarely such that a reliable reading to less than a scale division can be obtained.

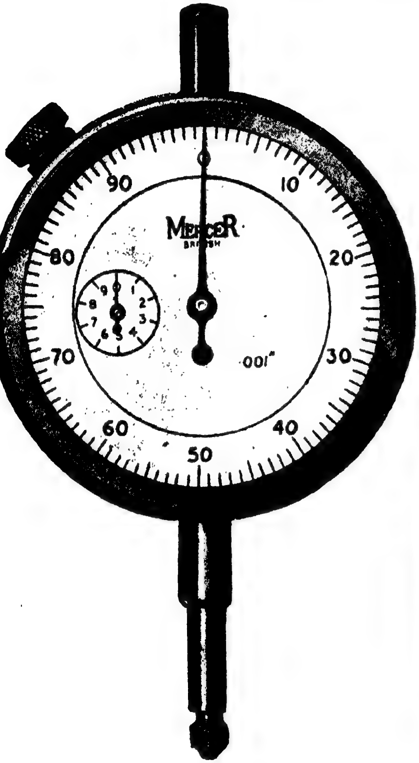


FIG. 17. THE DIAL GAUGE

FIG. 18. THE CYLINDER DIAL GAUGE

The dial gauge is used to measure flatness or axial truth when clamped on a block and stand, and used off a surface plate. With the component under measurement rotated in vee-blocks or centres, it is used to check concentricity.

Mounted on special extension handles it is used as a "cylinder gauge," and is the standard method of measuring long or large holes. Fig. 18 shows a typical gauge, which has three contact points, two close together spring loaded and opposite the third, which is adjustable. Between the two closely spaced contact points is a fourth plunger connected to the dial gauge. The gauge is

calibrated by means of a normal micrometer, and the error in setting is likely to be up to 0·0003 in. (7 microns). To read the gauge it is moved back and forwards so that the dial repeatedly reads a minimum, indicating the diameter of the cylinder in question. By a careful note of the average minimum reached by the needle, these gauges can be read to the nearest 0·0005 in. (10 microns) with accuracy, and are particularly useful in that they easily measure ovality or taper in a cylinder bore.

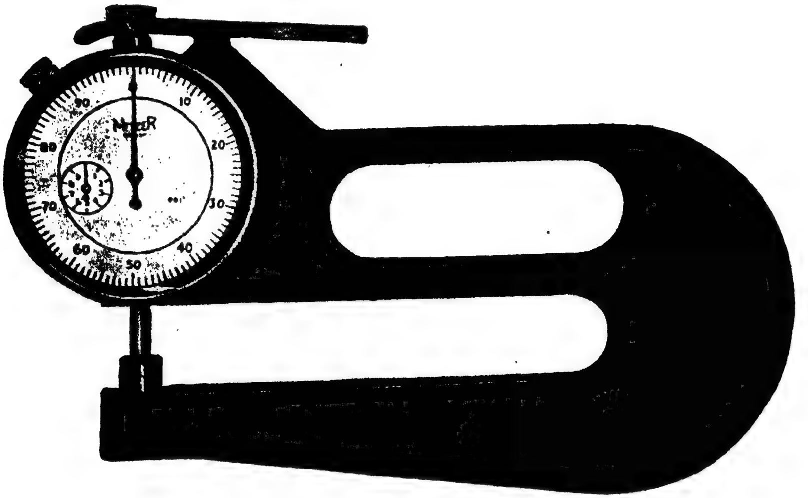


FIG. 19. THE DIAL MICROMETER

Dial gauges are also used in a form of micrometer (Fig. 19) particularly useful for the rapid measurement of sheets of rubber, leather, etc., or small components such as rollers, pins, etc., where great accuracy of measurement is not required. The constant measuring force of the dial gauge spring is an advantage in giving consistent readings but the method is sensitive to dirt and malalignment of the micrometer faces or sample being gauged, and an accuracy of greater than the nearest 0·001 in. (25 microns) is not to be relied upon.

On the other hand, in different form, the dial gauge has been effectively combined with a normal caliper snap gauge in the Zeiss Passmeter type of instrument (Fig. 20). In this case one anvil of the caliper is sprung and its movement recorded on a dial. An accuracy of 0·0001 in. (2½ microns) can be relied upon in absolute measurement when correctly set, and differences in size of half this can be recognized.

4. Precision Methods

(a) Lever Comparators

Comparators are similar to the dial caliper mentioned above in that they measure within small limits only when set to some absolute standard (usually slip blocks). They can thus measure within their own limits of accuracy plus the often greater limits of setting accuracy. (See Appendix 2.)

A comparator is essentially similar to a dial gauge except that the movement of the pointer is magnified many times by mechanical,

Fig. 20. The Zeiss Passmeter Type of Gauge

optical, or electrical means, usually by an optical pointer. Since this type of pointer can have a great effective length with correspondingly large lever magnification, the full scale movement of the pointer or light spot can correspond to extremely small movement of the plunger. Normal calibrations are thus: one division (about $\frac{1}{10}$ in.) equals from 0·001 in. for general "coarse" shop use to $\frac{1}{2} \times$ 0·0001 in. (1 micron). The instrument can therefore be read to about 0·00001 in. (10 micro-inches or $\frac{1}{5}$ micron).

Instruments of this latter accuracy are far too sensitive to be of much use for normal engineering component measurement, unless used in a controlled temperature gauge room and great care taken in the preparation and cleanliness of samples.

Such instruments, however, with the coarse graduation, are in wide scale use in the production measurement of rollers, gudgeon pins, centreless ground bolts, etc., which are ground to drawing tolerances of 0·0002–0·0003 in., and are satisfactory for use with unskilled female inspectors.

(b) Electrical Comparators

The Pratt & Whitney Electrolimit comparator (Fig. 21) uses a lever mechanism to unbalance an A.C. Wheatstone bridge, the movement of the lever being magnified on a sensitive micro-ammeter. An unusual feature is that the scale magnification is variable and must be set both as regards zero and some other dimension as well. While this is an advantage in many cases, it makes the instrument

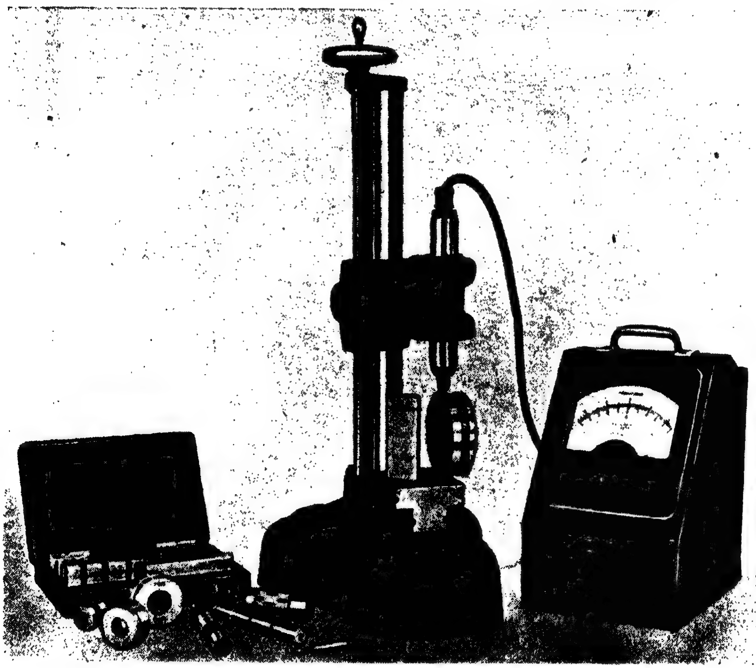


FIG. 21. THE ELECTROLIMIT COMPARATOR

more sensitive to maladjustment than one with fixed magnification. Internal and external holes can be measured, and a useful addition for measuring cylinder bores (above 2 in. dia.) is a plug type head connected to the instrument by a flexible lead.

The most sensitive scale calibration is 0·00005 in. (1 micron) per division, and estimation can be made to about one-fifth division (10 micro-inches or 0·2 micron).

(c) Pneumatic Comparators

The Solex gauge (Fig. 22) is a particular type of comparator depending on the back pressure of escaping air between the measuring

gauge and the sample. One or more holes in the measuring head allow the air to escape in the narrow space between them and the diameter or gap being measured; the back pressure being directly proportional to the gap, the magnitude of the latter can be

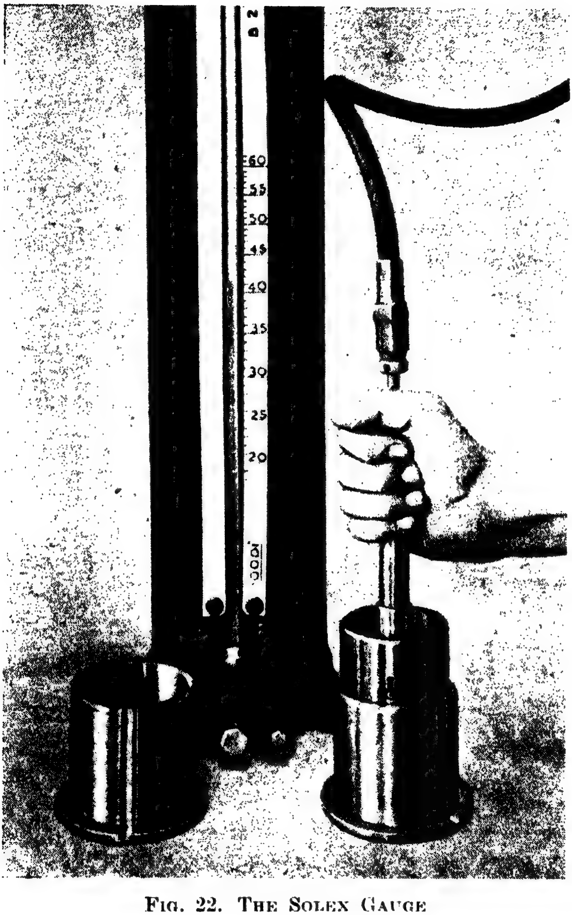


FIG. 22. THE SOLEX GAUGE

read off on a water level pressure gauge. Variations in air supply, etc., are all compensated for and the gauge is a well-tried and reliable instrument. With the normal scale division of 0·0001 in. the instrument can be used as a comparator to 0·00005 in. (or 1 micron), but as it has to be set to a ring or plug supplied by the makers, it is not considered quite as accurate nor as convenient

as some other types of comparator which can be checked against slip blocks.

A particularly interesting feature of the Solex gauge is that its head is smaller or larger than the diameter or gap being measured and thus need not scratch the surface as a plug gauge does, or a pointer of a dial gauge might; neither does it suffer from "brinelling" errors (see Appendix 2).

The normal type of measuring head has two opposite jets and will measure holes down to $\frac{3}{16}$ in. diameter, with a special type for

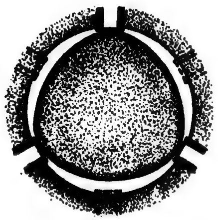


Fig. 23. The Detection of Lobing with a Solex Gauge

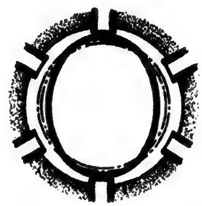


Fig. 24. The Measurement of Average Diameter with a Solex Gauge

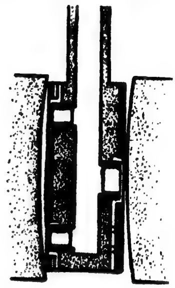


Fig. 25. The Detection of Axial Errors with a Solex Gauge

smaller holes. Blind holes can be measured to within about $\frac{1}{16}$ in. of the bottom.

The Solex gauge is unusual in that it can be made to detect errors of truth directly and is unique in its versatility in this respect. By using a head with three jets equally spaced, it can detect "lobing" or triangulation with constant diameter (see page 5) (Fig. 23). By using several jets it can indicate average diameter (Fig. 24), thus being particularly useful in such cases as tube drawing where the average diameter is more important than errors in ovality. By having three jets spaced apart in an axial direction (Fig. 25) errors in axial truth or curvature of a hole can be detected. Almost any measuring problem can be dealt with by suitably designed heads, including such difficult problems as the bore of a rifle.

A most interesting example of the extension of the Solex principle is shown in Figs. 26, 27, and 28. In this development, no fewer than six gauging points are operated at once, giving "Go" and "Not-Go" readings on six pressure gauges.

(d) *Slip Gauges or Blocks*

Slip blocks can hardly be considered as means of gauging any production component owing to the relative difficulty in using them

and their cost and delicacy. They are used universally as the standard reference in setting other types of gauge.

The designer should not specify tolerances which cannot be set directly by a set of slip blocks as this may involve a roundabout

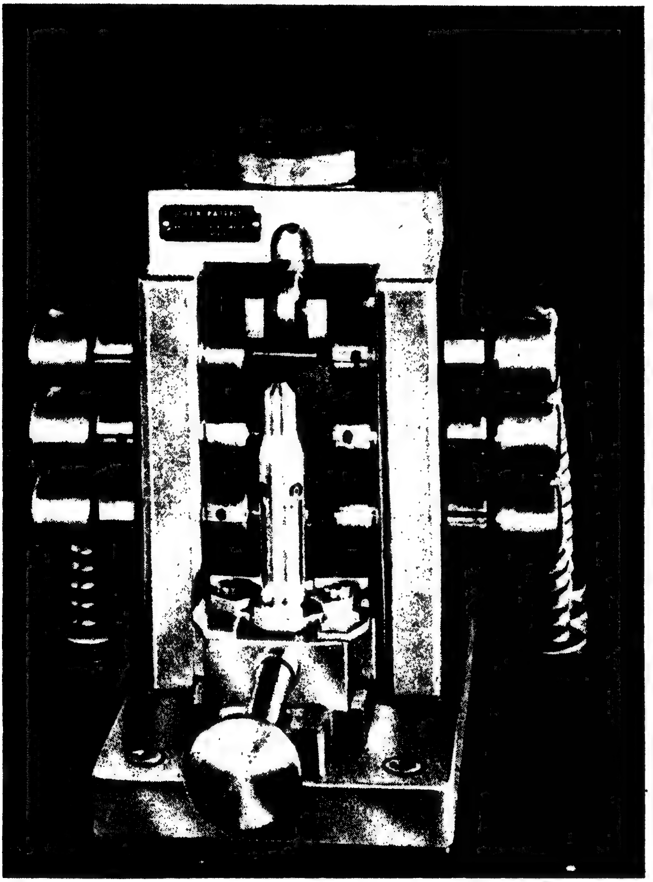


FIG. 26. CLOSE-UP VIEW OF A MULTIPLE GAUGING HEAD FOR 20 MM CANNON SHELL

method of setting, and anyway calls for a degree of accuracy unknown even in precision engineering. The normal slip block set (see B.S. 888) can supply multiples of 0·0001 in. or 0·0025 mm; an additional set supplies the two intermediate values needed with Newall limits (0·00025 in. and 0·00075 in.), and further sets can be used for multiples of 0·00005 in. or 0·001 mm (1 micron).

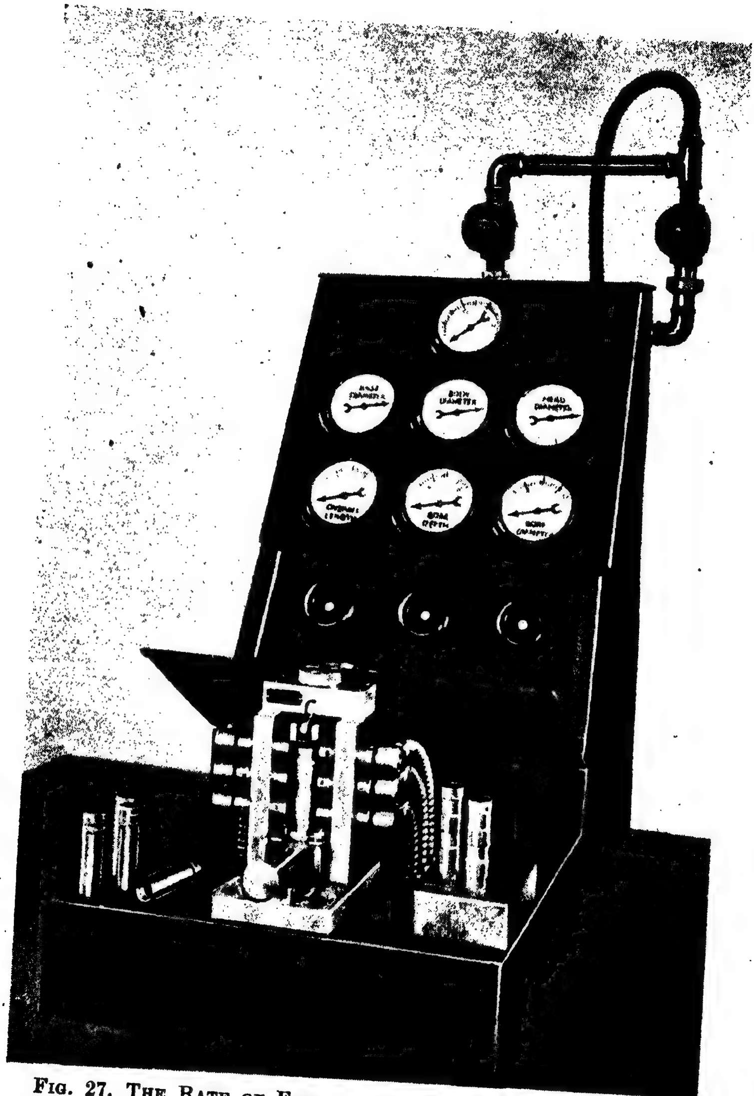


Fig. 27. The Rate of Flow of Air Through Each Orifice, or Valve, Governs the Reading of the Pressure Gauge, which gives a Measure of the Dimension

5. Miscellaneous Processes

(a) *Measuring Angles and Tapers*

The direct measurement of an angle on inspecting a component is difficult, and, as stated later in Chapter II should be avoided and the angle replaced by linear dimensions with suitable tolerances. When an angle must be specified it can be measured by a gauge with two angles corresponding to the limits of the angle quoted. This gauge is held to the part and an estimate of fit of the two parts is made. Since the eye can readily detect a gap of 0·001 in. if a strong light is held behind it, a very close estimate of the angle can be made if this light test can be used; and if the surface accuracy of the part is good. If the gauge is in contact over half an inch, a gap at one end of 0·001 in. corresponds to an angular difference of 1 in 500 or about $\frac{1}{10}$th of a degree. On the other hand, it would be virtually impossible to gauge a $\frac{1}{16}$ in. chamfer without optical projection to better than a few degrees.

Tapers are usually measured by sine-bars or by taper gauges which are "blued" to the part, two gauges giving the two limits. The blueing test is extremely accurate and will indicate clearly a surface deviation of less than 0·001 in. On a well-finished part a gauge will thus indicate the angle of the taper to less than $\frac{1}{10}$th of a degree.

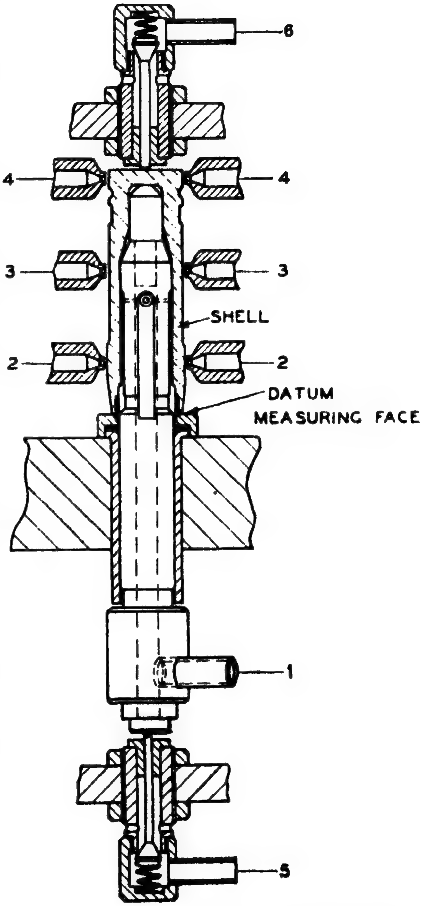


FIG. 28. THE SIX GAUGING POINTS: (1) FOR BORE DIAMETERS; (2), (3) AND (4) FOR EXTERNAL DIAMETERS; (5) FOR DEPTH OF BORE; AND (6) FOR OVERALL LENGTH

It is generally more convenient to specify a taper in terms of two diameters and the distance between them. Normal methods

of measurement can be used in this case, due allowance being made for the taper on the diameters being measured.

The measurement of the apex of a taper, or the position of some reference diameter can be done by means of a taper gauge, whose maximum diameter must not enter the taper by more than a specified amount. This means that the position of a particular diameter can be determined to within close limits, probably within 0·002 in.

A typical cone gauge is shown in Fig. 29.

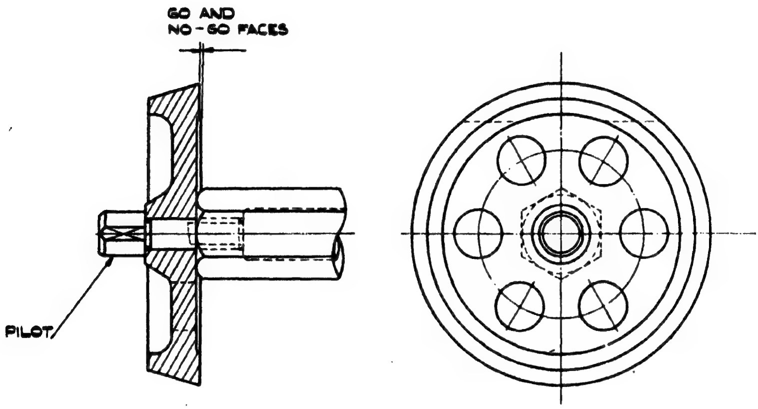


FIG. 29. A TYPICAL CONE GAUGE

Appendix 4 may be useful in converting angles into equivalent dimensions.

(b) *Measuring Hole Pitching*

The dimensions between the various holes on a component, not on a common axis, can be measured by fitting plugs in the holes and then measuring the linear dimension between the plugs by any normal method. However, this is rarely done on production components, as the holes are inevitably produced from a jig of known accuracy.

In applying tolerances to the position of holes, a proper knowledge of manufacturing limitations is more important than the method of measurement, and this will be dealt with later in Chapter II.

(c) *"Automatic" Inspection Methods*

An interesting inspection method is to combine several gauges or devices in one fixture so that a part can be fully inspected rapidly by trying it in the fixture or several stages of the fixture in succession.

Fig. 30 shows a typical set-up. While a detailed knowledge of these methods is not necessary to the proper use of engineering limits and tolerances, what a designing engineer must know is that there are often occasions when the best method of dimensioning and tolerancing a part is one which involves a special gauging tool.

For example, two parallel holes in a component might require special accuracy as regards mutual parallelism, and the engineer

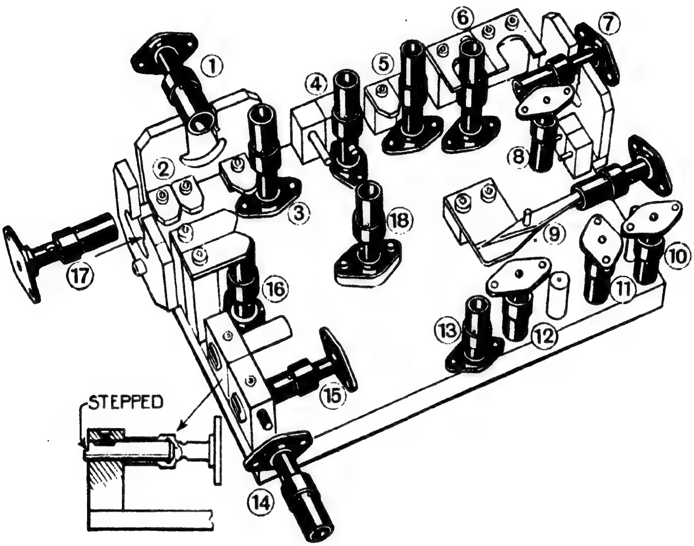


FIG. 30. A TYPICAL "AUTOMATIC" INSPECTION SET-UP
(Courtesy Aircraft Production)

might specify that the two holes are to be such that a pair of plug gauges fixed together with accurate alignment should be able to pass down the two holes.

This subject is briefly mentioned here and will be dealt with again in Chapter II.

(*d*) *Devices for Measuring Wall Thickness from One Side*

In the measurement of wall thicknesses, particularly on tubes and castings where it is not possible or practicable to get at the inside wall, measurement must be effected from the outside only.

In the case of castings, this measurement is often important to check core shift, and small holes are often permitted to be drilled so that the wall can be measured with a simple caliper made from a piece of bent wire. In many cases, such as on engine cylinder

block castings, hole drilling is not permitted and special devices have been developed.

These use electrical methods based on—

(i) The measurement of the resistance between two closely-spaced contact points pressed on to the outside of the metal section; used with a calibration reference piece, this method is accurate to about ± 10 per cent, but depends on the homogeneousness of the material.

(ii) The generation of eddy currents in the section, which must be magnetic and of a section not exceeding about ½ in.

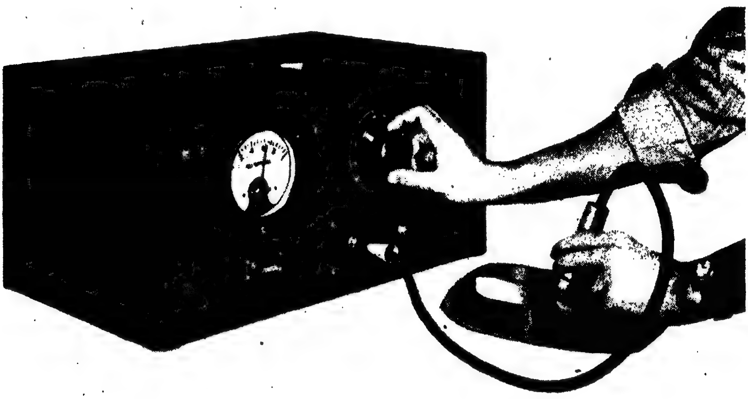


FIG. 31. THE SONIGAGE

(iii) A system measuring the time of passage of a supersonic wave from one surface, reflected back off the other. For any but thick sections, this involves very complicated and costly electronic time measuring equipment.

(iv) A newly developed system produced by the General Motors Corporation—the "Sonigage"—which measures the resonant frequency of the section and compares it with known samples with a claimed maximum error of 2 per cent. Sections of 0·02 in.–0·40 in. can be handled (Fig. 32).

It should be borne in mind that the provision of special equipment is a hardship to the factory and the designer should do what he can to avoid its necessity. It is usually possible on castings and tubes to permit tolerances which can be measured by quite crude caliper methods.

Further information on this subject is contained in a paper by Thornton in the *Proceedings of the Institution of Mechanical Engineers*, Vol. 140 (1938), p. 349.

6. Temperature Effects

All limits are intended to apply with parts and gauges at a temperature of 68° F. (20° C.). When gauging steel parts with steel gauges, variations of shop temperature are usually negligible, but when gauging light or special alloy parts, considerable errors may be introduced, particularly on long parts.

When the working temperature of an assembly is likely to differ greatly from 20° C., due allowance must be made for this in selecting limits. An interesting example of this is in precision high pressure gear pumps used on aircraft which have to operate at temperatures between + 70° C. and − 40° C., and involve clearance of 0·0005 in. between the gears. In these cases where light alloy is used, steel parts are made in special high expansion alloy.

When using heavy press or shrink fits, calculations often have to be made involving the coefficients of expansion of the materials.

Although coefficients of expansion are tabled in all engineering reference books, they are retabulated here again (Appendix 3), arranged in a form convenient for this particular purpose. The most general problem involves steel and aluminium, and it is convenient to memorize that "the differential expansion of aluminium and steel is approximately 0·001 in. per inch per 100° C."—i.e. 0·0008 in. per inch when put in boiling water.

TOLERANCES IN RELATION TO FACTORY MACHINE PROCESSES

1. Introduction

It is fair to say that almost any standard of accuracy of manufacture can be achieved with suitable equipment and given the necessary time. The methods adopted, for example, in the polishing of glass for lenses or the manufacture of slip blocks, produce surfaces of almost unbelievable accuracy. Table 2 is given hereunder for interest and indicates the manufacturing tolerances in micro-inches applicable to British Standard slip gauges (from B.S. 888).

It is absolutely wrong for an engineer concerned with normal or even precision products to contemplate limits or tolerances involving such accuracy even if it can be achieved, as the resultant product would be absurdly costly. He must consider carefully the economics of accuracy since it may be that his design will prove so costly to manufacture that it will have no practical appeal or value.

In general, it can be said that except for special cases in armaments, aircraft, etc., and for hand-made tool room or experimental shop work, the greater the quantity of production of a machine-made product, the greater the accuracy. This is because high production

TABLE 2

MANUFACTURING TOLERANCES OF SLIP GAUGES

(extracted from B.S. 888)

Unit = 1 micro-inch (0·001 × 0·001)

Size	Grade											
	Calibration			Inspection			Workshop A			Workshop B		
	Length	Flatness	Parallelism	Length	Flatness	Parallelism	Length	Flatness	Parallelism	Length	Flatness	Parallelism
0·01–1 in. (25 mm)	± 5	2	2	+ 7 − 3	5	5	+ 15 − 5	10	10	+ 25 − 5	10	10
1·0000–2 in. (50 mm)	± 10	3	4	+ 10 − 5	5	5	+ 20 − 10	10	10	+ 30 − 10	10	10
2·0000–3 in. (75 mm)	± 15	4	7	+ 15 − 8	7	7	+ 30 − 15	10	15	+ 40 − 15	10	15
3·0000–4 in. (100 mm)	± 20	4	7	+ 20 − 10	7	7	+ 40 − 20	10	15	+ 50 − 20	10	15

requires first-class machine equipment, which in turn can produce work of great accuracy without difficulty and with consequential savings elsewhere, as, for example, on assembly or in service.

With quantity production there is seldom much difficulty in achieving a necessary standard, but since by far the greater number of designers are concerned with batch or job production on equipment far from perfect, a detail study of manufacturing accuracy is necessary.

In the sections that follow, the problem is reviewed as comprehensively as possible, but so much depends on the particular circumstances existing in a works, the age and condition of plant, types of machine available, etc., that the designer can only be encouraged to investigate the problem on his own territory, preferably at first hand, by going into the shop and seeing for himself.

2. Interchangeability

Fits between one component and another are necessary whatever manufacturing methods may be used, but the desired result can be—and, of course, always was in earlier days—obtained by machining or fitting one part to suit the other. This practice is still common in such applications as big-end and crankshaft bearings on internal combustion engines, compressors, etc. Another equivalent process in extreme precision engineering is the lapping of a plunger into a bore, as in the case of a Diesel injector pump or a hydraulic slide selector, where the required fit is too fine to be obtained directly from a machine process.

Except in the unusual event of a component which is to be machined to fit some other component, all dimensions determining a part must have limits, however wide, to enable it to be made and inspected to the drawing.

The most important function of limits is, however, to enable interchangeable components to be produced. The advantages of interchangeability, in the factory in reducing assembly time, and in the field or in service in facilitating repair or replacement of worn or defective components, are well known and universally accepted. Interchangeable manufacture is standard in quantity produced articles, and great strides were made during the War in extending its principles to such large products as aircraft and in some degree to ships.

In many cases full interchangeability cannot always be achieved, but compromise is often perfectly satisfactory. Spare pistons and piston rings on motor car engines, for example, have usually to be selected from a small range. Fitting adjustments are often eliminated by shims and the like, which have to be added during the fitment

of a spare part until a particular adjustment is obtained, but again the supply of shims is usually organized to facilitate this. Even on heavy machinery such as large Diesel engines a considerable degree of interchangeability is usually achieved.

It is important to appreciate that interchangeable manufacture almost invariably benefits everyone, and by suitable detail design can usually be adopted. Where it is better that a component is not "broken down" into individual parts each one interchangeable, partial "break down" into interchangeable sub-assemblies is desirable.

3. Tolerance and Limits

As explained previously, the designer determines the required fit of two components in terms of the *basic deviation*, which determines the minimum clearance or interference. The manufacturing tolerance allowed to the factory worsens the fit and should be as wide as can be permitted. In the case of non-mating parts or dimensions, wide general tolerances are applied to control the shape, weight, or other aspects of the part, and to prevent fouling, etc., of other components in the assembly.

General limits are therefore required to control the general aspects of the specification.

Special limits are necessary with interchangeable manufacture—

(i) to control the fits between mating components; and
(ii) to maintain desired clearances.

(a) *General Limits of Tolerance*

Normal general engineering practice is to apply a general tolerance of $\pm \frac{1}{64}$ in. ($\pm$ 0·015 in. = $\pm$ 0·4 mm) on all non-limited dimensions.

On aircraft and equivalent work, the standard tolerance is $\pm$ 0·010 in. (= $\pm$ 0·25 mm) for machined parts, and on aircraft sheet metal parts

$$\begin{array}{l} +\ 0\cdot02\text{ in. }(+\ 0\cdot5\text{ mm}) \\ +\ 0\cdot00\text{ in. }(+\ 0\cdot0\text{ mm}) \end{array}$$

For more precise parts and particularly small ones $\pm$ 0·005 ($\pm$ 0·15 mm) is frequently used, but is rarely justified as a general tolerance, except on small instrument components.

(b) *Quality of Tolerance or Grade of Workmanship*

In the I.S.A. tolerance system described in Chapter IV, a system of tolerance gradation is used which makes a convenient method of classifying the relative severity of tolerances.

Fig. 32 indicates the system and the tolerances involved graphically. The numerical qualities are used later on in this section in

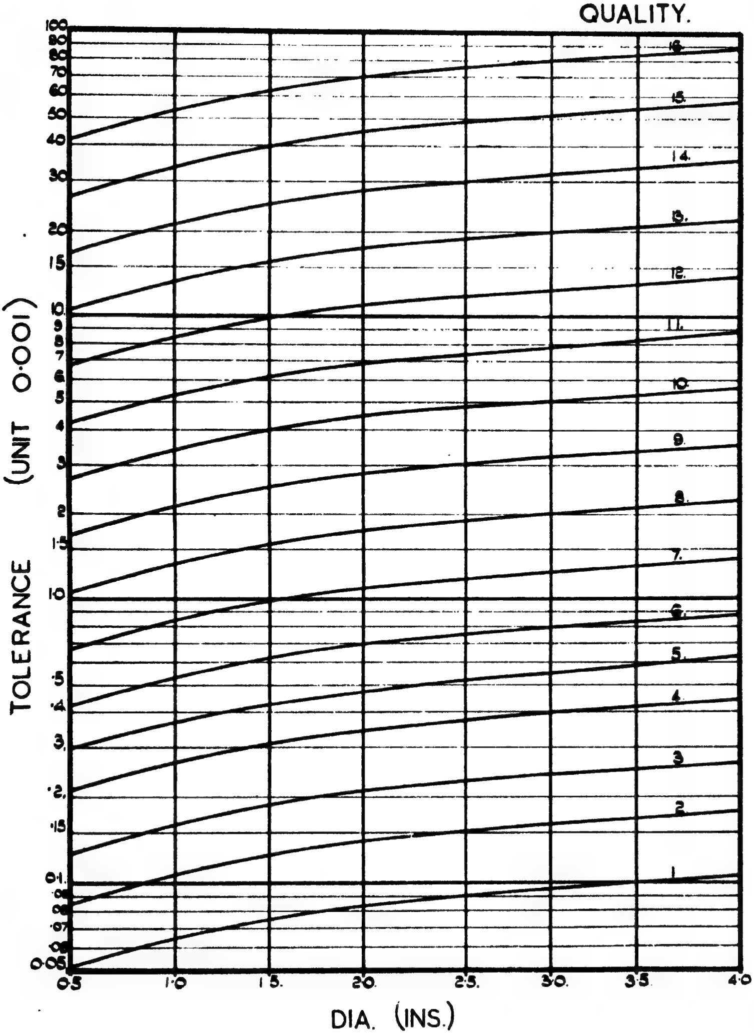


Fig. 32. The Tolerance Qualities of the I.S.A. Tolerance System

considering the possible accuracy which can be achieved with different machines or other processes. Table 3 gives the same information in tabular form with an indication of the appropriate

TABLE 3

Working Tolerances on Various Machine Processes

I.S.A. Quality	Relative Magnitude	Actual Tolerance on Dia. (Unit 0·001)				Intended for	Applicable to these Components or Machines
		0·5 in.	1 in.	2 in.	4 in.		
1	1·2	0·0505	0·063	0·0815	0·103	Gauges	Slip blocks.
2	2·0	0·084	0·106	0·136	0·172	Gauges	High quality gauges.
3	3	0·126	0·160	0·204	0·258	Gauges	Good quality gauges.
4	5	0·210	0·265	0·340	0·430	Gauges	Medium quality gauges.
5	7	0·294	0·371	0·475	0·6025	Gauges	Ball bearings; machine lapping; diamond boring, fine grinding.
6	10	0·420	0·530	0·680	0·860	Gauges / Fits	Grinding, fine honing.
7	16	0·673	0·850	1·087	1·375	Gauges / Fits	High quality turning, broaching, honing.
8	25	1·050	1·325	1·701	2·150	Fits	Centre lathe turning and boring, reaming, capstan or automatic in good condition.
9	40	1·68	2·12	2·72	3·44	Fits	Worn capstan or automatic; horizontal or vertical boring machine.
10	64	2·68	3·39	4·42	5·5	Fits	Milling, slotting, planing, metal rolling or extrusion.
11	100	4·2	5·3	6·8	8·6	Fits	Drilling, rough turning and boring; precision tube drawing.
12	160	6·73	8·475	10·9	13·8	Not for fits	Light press work; tube drawing.
13	250	10·5	13·25	17·01	21·5	Not for fits	Press work; tube rolling.
14	400	16·8	21·2	27·2	34·4	Not for fits	Die casting or moulding; rubber moulding.
15	640	26·8	33·9	44·2	55·0	Not for fits	Stamping (approximately).
16	1000	42·0	53·0	68·0	86·0	Not for fits	Sand casting (approximately); flame cutting.

(Tolerance $= 0{\cdot}52\sqrt[3]{D} + 0{\cdot}001D$)

processes which can work to these tolerances. Although these are by no means the limit of accuracy of the processes referred to, they do represent the minimum tolerances which can be given to the various processes for economical production and designers should always bear this in mind. The various processes are dealt with in detail in subsequent pages.

4. Limits and the Production Facilities

It is useless for a designer to ignore the facilities available in his factory for achieving machine or process accuracy when tolerancing a drawing. It is useless, for example, in a small factory producing large air compressors to call for interchangeable limits on the crank-shaft and big-end bearings, when precision grinding and boring machines are not available and the only equipment is a crank pin turning lathe and a centre lathe for boring the bearing shells for subsequent scraping.

It must not be thought, however, that the lack of precision equipment means that limits must be expressed in coarse or round figures, e.g. multiples of 0·001. The accuracy of measurement is in general independent of the dimension to be measured; there is no reason to think that limits of 0·001/0·002 in. can be measured any more easily than 0·0012/0·0024 in., and the fact that a limit may be expressed in 0·0001 in. or even 0·00005 in. units has no connection with the type of machine to be used in its achievement. The significant point has already been considered, i.e. that it is the measuring or gauging equipment available which determines the logical fineness of tolerance units.

Theoretical limits are generally rounded off to about 10 per cent so that the finer the limit, the greater the accuracy of expressing it; to round off 0·0094 in. to 0·009 in. is perfectly logical, but 0·00034 in. should be 0·00035 in., not 0·0003 in.

It is preferable to have a given system of tolerances and to use the tabled figures even in the case of a product made with inaccurate equipment, rather than to endeavour to round off the limits to more than about 10 per cent, as this involves special tables and non-standard tolerances. Many practical engineers in job production shops, if confronted with a dimension of $\frac{17}{64}$ in. – 0·00045/– 0·00125 would say that the fifth decimal place in the limits was absurd. This, however, is not so, since the nominal dimension itself involves *six* decimal places ($\frac{17}{64}$ in. = 0·265625 in.). The logical procedure is to use tables of tolerances determined on sound theoretical premises and suitable for precision or non-precision manufacture, choosing from the tables to suit the production set-up, and to ignore the last decimal places of the *size* (not the limit alone)

as necessary. An example of this will make the matter more clear—

$$\tfrac{17}{64}\text{ in.} = 0{\cdot}265625\text{ in.}$$

Limits	— 0·00045/— 0·00125 in.
Actual size	0·265175/0·264375
Rounded off to	0·2652/0·2644

If the limits alone had been rounded off to

—.0·0005/— 0·0012

The actual size would have been

— 0·265125/— 0·264425

which is no better than before and would still have to be rounded off again.

On the other hand, the designer must admit that he is often faced with prejudice against what may appear to the engineer, who prides himself as being "practical," as unnecessary and unjustified fineness of expression in using five or even four decimal place inch tolerances.

Care and tact are necessary because the designer, although correct, may annoy and upset the practical man, and lead to the unsatisfactory state of affairs often found in a works where the Works and Inspection Departments use "discretionary" powers and have their own ideas on interpreting drawing limits.

This state of affairs is usually made worse by the designers specifying tolerances which are too severe, and by the Production Department finding out by experience that wider limits perform adequately in the field, and therefore taking upon themselves these discretionary powers.

A much better state of affairs occurs when the production engineer comes to the designer and actually proposes that as he is now in a position to produce with greater accuracy, he can accept tighter limits. If the designer will adopt a reasonable and sensible attitude towards the limits and not surround them with an air of mystery, a spirit of mutual confidence will be built up between him and the shops, where the drawing is worked to and discretionary powers are not known.

5. Machine Limitations

In order to apply limits satisfactorily, detailed knowledge of the working accuracy of various machines and methods is necessary, as mentioned previously. In the following sections the various common machine processes are considered in relation to dimensional accuracy and their sources of error discussed. The qualities of Fig. 32

are referred to throughout. Other fabrication processes are considered in Chapter VI.

(*a*) *Lathes*

Centre Lathes. Almost any degree of accuracy is obviously obtainable on a centre lathe, depending only on the quality of machine and skill of the operator. Centre lathes in the tool room produce work of very high quality.

Capstans and Turrets. In some respects these may be equivalent to centre lathes, in that good work can be produced with skilled operators. However, they are essentially machines for semi- or unskilled operators, and the accuracy achievable usually is limited by the machine. Long diameters are turned with roller boxes, and errors may be due, for example, to variation in cutting feed controlled by the operator (excessive feed produces oversize work), to malalignment of the roller box and the spindle, or to tool spring. In addition, adequate allowance must be made for tool wear and the error of setting up the roller box. Quality 8 or 9 tolerances can therefore be reasonably called for, for roller box work. The slide rest is often used for turning, but here the stops on the cross traverse introduce errors, unless handled with special care; a similar degree of accuracy can be expected. Holes are drilled or reamed using tools in the turret, and Qualities 11 and 8 respectively can be expected.

Lengths are sometimes determined by the turret stops, and the care and evenness of manual pressure of the operator affect results. On collet work there is a tendency for the bar to slip during cutting. Quality 10 is as much as can be expected.

Automatics. These are similar to capstans except that as cams replace the stops and no operation can be done too fast, the quality of work is often better than on a capstan. Special processes have been developed for the reproduction in quality of small precision parts turned to Quality 7 or 8 limits, using sizing tools, but detailed study is needed of the machines available and the particular component to be produced to get the best results. Bar slip in collet work is, if anything, worse than on a capstan, where the operator usually exerts considerable clamping effort.

Semi-automatic lathes for large components can work to similar tolerances. With special boring tooling, ball race fits can be bored on cast iron or steel housings.

(*b*) *Milling Processes*

Milling. Milling itself is not an accurate process, probably due to the large number of teeth cutting at the same time and the distortion introduced by the appreciable cutting forces used. Other

dimensional errors are introduced by the cutters not running true. The milling machine itself can be as accurate as a lathe when used for boring, etc., with single point tools, but in its normal duty, Quality 9 tolerances are all that can be expected. More accurate work requires extra finishing cuts and careful setting.

Apart from the error in the width of a slot milled by a cutter, or the width milled by a gang of cutters, the setting of the cutter or cutters in relation to the work in its jig on the machine table introduces other errors. Thus, the accuracy of setting the cutter in relation to the component can be taken as the same as on the dimension being milled (i.e. Quality 9); greater latitude (Quality 10) should be allowed if possible to allow for simpler setting devices on the jig.

Right angle axis errors, i.e. errors on the squareness of milling in relation to the vertical or horizontal axis of the components, depend on the type of jig used, but an accuracy of 0·001 in. per inch or about 3 minutes of an arc (0·05°) can reasonably be expected. Errors of alignment in the direction of movement of the table are much smaller.

Angle division or indexing errors, as when milling gear teeth or splines, are due mostly to dividing head faults. With a dividing head in good condition, an accuracy of 1 or 2 minutes of an arc can be achieved. When milling splines and the like, angle and width errors are both present and the position of one side face of slot or tongue in relation to its nominal position to less than Quality 8 cannot be expected, the tolerance being applied to the width being milled. This question is discussed in greater detail in Chapter IV.

Keyway slots (Woodruff or rectangular keys) can be milled with small cutters to Quality 8 tolerance as regards width.

HOBBING. Hobbing, thread milling, gear shaving and equivalent processes are more accurate than milling as the forces involved are generally smaller and the accuracy of reproduction of the hob-generated shape is high; profile accuracy to about Quality 7 can probably be achieved, but much depends on the shape of the hob, condition of the machine, etc. Hobbing is usually used on processes such as gear cutting where tolerances are special and not normally specified by the designer except by implication in terms of gear backlash, noise, etc.

(c) *Rectilinear Processes*

Under this heading is the work of shaping, slotting, and broaching machines (and strictly planers, dealt with below) where the tool moves rectilinearly during cutting. Shaping and slotting are not particularly accurate because of their relative coarseness of feed

and the consequent unevenness of surface. Milling tolerances can be worked to, although shallow slots can be made by a skilled operator to Quality 7 tolerances. For production work much of the work done on shaping or slotting machines is done on a broaching machine, either hole broaching or surface broaching.

Owing to the slow, steady, burnishing action of the broach, remarkable accuracy is possible on splined or keywayed holes and irregular profiles in the case of surface broaching. Hole diameters and slot widths to Quality 6 or 7 are possible with hole broaches, and in the case of surface broaching, profile accuracy corresponding to Quality 7 or 8.

Gear shaping with special machines can be done with great accuracy but the remarks on hobbing above apply.

Surface grinding, which might be considered as a rectilinear process, is discussed under the next heading.

(d) Abrasive Processes

Grinding. External grinding is remarkably accurate and Quality 6 is possible on repetition work, both on centres or centreless. Special automatic and continuous size reading attachments (Fig. 33) are available, enabling work to Quality 5 to be produced in large-scale production.

Internal grinding is as accurate as external work (Quality 6) when the spindles involved are short and rigid. With tube grinding, long spindles are necessary and Quality 7 or even 8 is as much as can be expected owing to whip of the spindle, chatter, etc.

Surface grinding can reproduce with accuracy depending almost only on the slides of the machine, and certainly to Quality 3 or 4. Location errors in holding the work may be greater than grinding errors.

Honing. Honing can produce work of equal accuracy (Quality 6) to grinding on short holes and with greater facility. On long holes, it is better than grinding and, in fact, can produce holes as regards diameter errors to Quality 7 about 24 in. long with a surface finish equal to a mirror, although axial truth will not be so good as this.

Honing is an effective replacement for reaming on holes in steel, particularly when press fit bushes are used as accuracy is achieved cheaply and with the good finish required for press fits.

Lapping. Machine lapping can produce cylindrical parts to Quality 4 tolerances without difficulty. A typical example of this is on piston gudgeon pins. High pressure hydraulic pump pistons (see Fig. 63) are machine lapped to Quality 2 tolerance without difficulty.

(e) *Boring Operations*

DRILLING. A drill will cut oversize if the point is not ground perfectly true and central and will produce a very rough hole if the feed and coolant are not carefully regulated. Drilling is an indispensable process, however, and Quality 12 tolerances applied unilaterally in an oversize direction can be worked to.

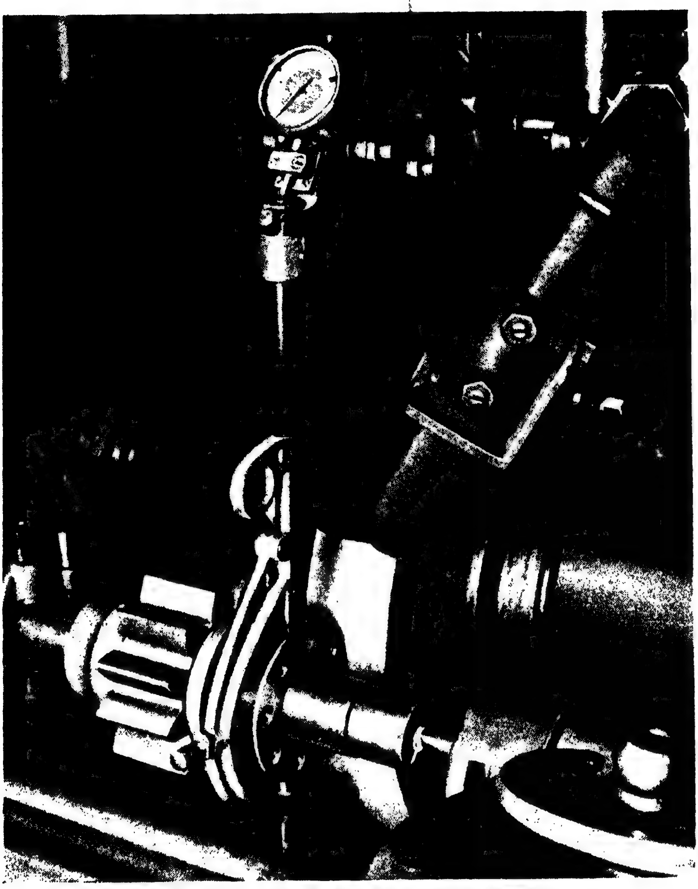


FIG. 33. A GRINDING SIZING ATTACHMENT

When drilling long holes considerable wander may be expected; when drilling a $\frac{1}{4}$ in. hole through 6 in. of steel the wander may be as much as 0·06 in. and designers should note this when designing long holes with high length/diameter ratios (e.g. crankshaft oil passages).

Reaming. Reaming is usually carried out direct from a drilled hole up to about $\frac{3}{4}$ in., and above this from a bored hole. A reamer can be made to cut appreciably oversize or very slightly undersize, depending on its keenness of edge and the way it is used. Quality 7 tolerances are reasonable, but involve careful selection of the reamer. British standard reamers are made slightly oversize so that they tend to cut large, thus extending their life. A new standard reamer may therefore cut above normal unilateral hole limits.

Boring. Single point tool boring accuracy on a lathe is a question of the skill of the operator and the condition of the machine. Boring with boring bars with accurately set tools can by suitable combinations of roughing and finishing cuts, produce Quality 7 holes. Special types of reamers or boring bars for small holes can produce Quality 7 holes down to about $\frac{1}{2}$ in. diameter and Quality 6 down to about $\frac{3}{16}$ in. in lengths up to about $10D$. Rifle, gun, or tube boring, involves special equipment on which the accuracy will depend, but the errors are more of an axial nature than on the diameter.

Fine boring, using special precision boring machines and diamond or tungsten carbide tools, can produce holes to Quality 5 tolerances or even Quality 4 on short holes of ample diameter.

Adequate provision for substantial spindles must be made if full advantage of these excellent machines is to be taken, and the designer can often help here by bearing this in mind. Fine boring on a high-speed lathe is not the same thing, as the work rotates and is supported on spindle bearings in nothing like as precise a manner as the boring bar of the special boring machine.

(f) *Heavy Machines*

Under this heading are included planing machines, horizontal and vertical boring machines, heavy lathes, and the like. Accuracy depends almost entirely on the skill of the operator and the quality of the machine, but each type has obvious limitations. It is useless to try to bore a long hole on a horizontal boring machine to less than Quality 9 tolerances (hard materials) or Quality 8 (soft materials), as the boring bar will whip appreciably. On the other hand, Quality 7 holes can be produced on a vertical boring machine with correctly set boring bars on a short bore, as on a cast-iron flywheel.

A plano miller as used for aircraft spar milling may be able to mill a width to Quality 8 tolerances locally, but errors of straightness would be considerably in excess of this.

Fortunately, excessive accuracy is rarely expected in the type of component produced on heavy machines and the design is such that unreasonable limits are not used.

CHAPTER II

LIMITS AND FITS IN THE DESIGN OFFICE

INTRODUCTORY

THE determination of tolerances is an extremely skilled job and requires knowledge of their significance, appreciation of their difficulty of realization in the shops, and in particular, an understanding of their effect on the finished product both as regards initial performance and wear in service.

In far too many design offices are tolerances and limits selected by juniors from convenient engineering reference books, without the proper study of the product by engineers specializing in tolerance problems. During the war hundreds, perhaps thousands, of sets of drawings were sent out to sub-contracting firms for complete manufacture of a product where the new firm had been able to point out that the limits of many parts were absurd, or would not permit assembly under adverse conditions. Usually, if the design was not new, the parent design firm knew of the faults in their drawings and had made production adjustments without bothering to alter the drawings. These faults occur owing to inadequate understanding of the problem or inadequate study of the design.

In general, the larger manufacturing firms producing equipment in quantity have learnt by their own mistakes in the past and do apply the necessary specialist study to their products; smaller and less experienced firms can do much by suitable study.

It cannot be too strongly urged that all design offices should be given selection tables of as detailed and specific a nature as possible to work from, and that they should not be allowed to work directly from the comprehensive B.S.I. or I.S.A. systems. The chief engineer should specify in the selection table the type of fit for all jobs in the firm's class of product in terms of the tolerance system reference symbols, and see that individual designers work in terms of this classified reference when discussing or considering fits, only putting on the actual limits on the drawing once the fit is established. Special fits, when necessary, should be allocated by senior engineers. It is better for the mental process of the designer to be as follows—

> "This is a normal turning fit shaft in a hole and an average good fit is required. My table says 'H8–f7' for this. . . ."

than to be—

> "I want about 0·001 minimum clearance and don't mind a few extra thou. Let's try an 'H8' hole and probably an 'f8' shaft . . .; no, that won't do, an 'f7' appears to be the correct mate for an 'H8' hole. . . ."

Questions of clearance fits involve lubrication and similar problems; interference fits involve stressing the components, problems of assembly, temperature effects and the like, and only a senior engineer can be expected to have the necessary experience to make a decision. There is very little risk that once a type of fit has been determined for a type of duty, changes in diameter will upset the choice, because the standard tables of limits take this into account; where a change of fit as a diameter varies is required this can be indicated on the selection table.

The use of selection tables and the eliminations of choice from the ordinary designer invariably means better tolerancing from the production engineer's point of view. The average designer rarely appreciates the significance of clearance and usually will, without guidance, ask for too accurate a running tolerance. Conversely, he is usually unaware of the difficulties of press fits and without expert help would often specify a press fit which was too slack or too tight on extreme limits.

Although it has become a platitude to say that designers are always asking for excessively tight tolerances, it is true to say that this is often the case, and that a better comprehension by them of the significance of tolerances would prove beneficial to the production engineer.

In the sections that follow, as much advice as possible will be given on the problem as well as on the preparation of selection tables.

Table 4 shows a particular selection table in actual use in one drawing office, and is of the type that can be strongly recommended. Another more general selection table for S.K.F. ball bearings is given in Chapter V.

DRAWING PRACTICE

1. General Tolerances

These have already been mentioned in Chapter I, page 30, as—

General engineering	± 0·015 in.	± 0·4 mm
High-class engineering	± 0·010 in.	± 0·25 mm
Extreme precision and instrument engineering	± 0·005 in.	± 0·15 mm

For other coarse tolerances, the following divisions of the inch

TABLE 4

Typical Tolerance Selection Table

HOLES	
H7	Bore produced on lathe.
H8	Bore produced on lathe or boring m/c.
H9	Bore for parts fitted with rubber rings where fine fit is not essential.
D8	Bore for sh. abs. bottom bushes (to produce sliding fit with p6 shaft) and diaphragm (to produce clearance with H9 shaft).

SHAFTS

Group	Designation	Use with Hole	Description	Examples
Press fits	p6	H7	Press fit	Sh. abs. pistons and end caps—bushes pressed into torque link lugs and levers (re-reamed afterwards).
	s7	H8	Press fit, to be used where bore is produced on boring m/c for light alloys only	Bushes in large light alloy castings and forgings (re-ream after pressing in).
Push fits	k7	H8	Light press fit (distortionless)	Bolts in bushes which rotate on O/D (torque links, selector levers, etc.)—needle races.
	j6	H7	Precision push fit	Spigots on eng. driv. pump parts and other components requiring high accuracy.
	j7	H8		All furnace brazed parts.
Locating fits	h8(1)	H8(2)	Dismantleable spigot	Jack end fittings—lock housings.
	f7	H7	Locating fit for threaded parts	Piston rod plugs if carrying piston—bushes screwed into shock absorbers.
Running fits	g6	H7	Fine sliding fit	Glandless pistons which must achieve a measure of sealing—e.g. on shuttle valves, pneumatic jacks, etc.
	f8	H8	Sliding or turning fit	Bushes—sliding tubes—guides—sliding assemblies not fitted with rubber rings.
		H9		Pistons fitted with rubber rings if (*a*) piston moves or (*b*) very accurate location is not required—moving shafts passing through glands (except chromed piston rods in small sizes).
	f9	(3)		Flash chromed jack piston rods—small sizes.
	p6	D8	Sliding fit	Sh. abs. piston tubes running in bottom bush.
Free fits	e9	H8	Clearance fit	
	h9	D8	Clearance fit	Sh. abs. piston rod in diaphragm.

(1) h9 on sh. abs. piston rods.
(2) H9 may be used if adequate for other considerations (e.g. jack and shock absorber cylinders).
(3) H9 on gland carrier—H8 on bush.

WIDTH AND LENGTH FITS	
H9	Width inside fork ends—length of shouldered bolts.
k9	Clamped bushes rotating on O/D, e.g. in torque links, selector and hand pump levers, etc.
h9	Width over non-moving lugs.
e9	Width over lugs rotating in fork (unless on clamped bush)—length of bushes if not clamped.
d9	Width of lever or lug rotating on clamped bushes.

or millimetre should be used in place of fractional or other dimensions—

Inch: 0·1, 0·05, 0·02, 0·01, 0·005
mm: 1, 0·5, 0·25, 0·1

2. Method of Indicating Limits

Various systems are common for indicating limits, as follows—

System A. ″·374/″·373
System B. $''{\cdot}374_{-1}$
System C. $''{\cdot}375\begin{smallmatrix}-0{\cdot}001\\-0{\cdot}002\end{smallmatrix}$ or $\frac{3}{8}''\begin{smallmatrix}-0{\cdot}001\\-0{\cdot}002\end{smallmatrix}$ or $10\text{ mm}\begin{smallmatrix}-0{\cdot}020\\-0{\cdot}043\end{smallmatrix}$

System A is an attempt at realism by specifying the actual sizes to be measured to save the operator or inspector having to add or subtract. The operator, however, usually works with a caliper or plug gauge, and is not concerned with the actual size. Furthermore, this system hides the nominal size so useful in asking for or finding a gauge, particularly with decimal equivalents of fractional inch sizes. This method is not so readily applicable to the metric system where one of the advantages is in the fewer number of figures used owing to the better size of the basic unit.

System B is an attempt at specifying the *basic deviation* by giving the size the designer really wants and then the tolerance or allowance away from this. It appears attractive when the limits concerned are multiples of 0·001, but is less attractive when standard tables are used involving multiples of 0·0001 in. or even 0·00005 in. It has the same objection as with System A that the nominal size has to be worked out. It is equally inapplicable to the metric system.

System C is to be preferred to all others because—

(i) it is applicable to inch or metric systems;
(ii) it specifies the nominal size;
(iii) the type of fit is readily estimable; in fact, the actual symbol reference is often added to facilitate obtaining the correct gauge; e.g. $1{\cdot}0\begin{smallmatrix}+0{\cdot}001\\+0\end{smallmatrix}$ in. "U";
(iv) it encourages engineers and inspectors to be classification conscious and assists in spotting mistakes, as similar numbers repeat frequently and often many limits can be memorized;
(v) on the other hand, the operator does not have to calculate the size, as he works to a set gauge;
(vi) limits can be transferred from tables directly without subtraction or addition.

This system is used throughout this book.

Opinions and practice differ on the question of the sign to be given to the zero in unilateral limits. Preferred practice is to give the zero the same sign as the other limit—e.g.

$$1{\cdot}125\,{}^{+\,0{\cdot}001}_{+\,0} \quad \text{or} \quad 1{\cdot}125\,{}^{-\,0}_{-\,0{\cdot}002}$$

in each case the zero being put in its correct position in relation to the other limit.

Other systems give

$$1{\cdot}125\,{}^{+\,0{\cdot}001}_{-\,0} \quad \text{or} \quad 1{\cdot}125\,{}^{+\,0}_{-\,0{\cdot}002}$$

and

$$1{\cdot}125\,{}^{+\,0{\cdot}001}_{0} \quad \text{or} \quad 1{\cdot}125\,{}^{0}_{-\,0{\cdot}002}$$

The first-mentioned practice is preferred because if any mistake is to be made in reading the zero limit it will be in a safe direction. The usual English method ${}^{+\,0{\cdot}001}_{-\,0}$ is bad in that it is often misread for $\pm\,0{\cdot}001$ or ${}^{+\,0}_{-\,0{\cdot}001}$.

In the preferred method, there are always two correct signs which can hardly be misread. This is merely a convenient convention as $+\,0$ or $-\,0$ does not mean anything. The zero without sign is actually the most logical.

3. Choice of Tolerance System

(a) Standard Systems

There are several standard systems of limits and tolerances which the designer may use and full details and tables are given in Chapter V. These systems are—

THE NEWALL. This is the earliest of all the systems and although its extreme simplicity makes it appear attractive, this feature is actually the main reason for its being obsolescent. The holes in the system are bilateral; there are too few fits given and those that are listed do not conform to modern ideas as regards their basic deviations. This system has served a useful purpose in the past but is not suitable for modern production and should not continue to be used by designers.

THE B.S.I. (B.S. 164). This system, although listing bilateral holes, recommends a series of unilateral holes and a full range of shafts to be used in conjunction with them. The main criticism of the system is that although there are four qualities of hole, there is only one quality of shaft in each fit, and although the tolerances

on these fits are reasonable for some classes of work, for ordinary general engineering use they are too tight. In a better system more than one quality of tolerance is listed for most classes of fit. There are other criticisms that can be made of the tolerance units, diameter increments, etc. (some of which are dealt with in Chapter III), but the system is useful where a British Standard system is preferred. The standard is under revision by a Sub-committee of the B.S.I. and a revised standard is in preparation at the time of writing.

THE A.S.A. SYSTEM. The American Standard system is little better than the Newall in that too few fits are tabled and its revision is in hand by the A.S.A. A single specified hole and shaft are given in each of the various fits; all holes are unilateral.

THE D.I.N. SYSTEM. This German metric system is almost identical with the I.S.A. system, and no reader of this book is likely to want to apply it instead of the I.S.A. system; information on it is given for reference only.

THE I.S.A. SYSTEM. The I.S.A. system is by far the most comprehensive and detailed of all tolerance systems; it is also the most modern. It lists a full range of 21 holes and 21 shafts covering all possible requirements; it gives several qualities of tolerance for each grade of shaft or hole; it lists all these possible combinations and recommends for general use a few unilateral holes and an adequate range of shafts, the remainder being listed for special or particular duty (e.g. oversize holes). The system lists what all others should do—full details of gauge tolerances.

The original I.S.A. system publication is in metric units. The American Society of Mechanical Engineers published an inch conversion in 1942, but, as explained later in Chapter V, this was not done correctly; the author in conjunction with two colleagues has translated the whole work and produced inch tables which are identical in significance with the metric original but convenient for the inch system as regards tolerance units, diameter steps, etc. These tables are reproduced later on.

Although the author may be accused of personal bias in view of this, he holds the strong opinion that the I.S.A. system is vastly superior to all others and believes that any designer who takes the trouble to study it will agree.

(b) *Unilateral or Bilateral*

All modern tolerance systems recommend unilateral tolerances on the hole basis, i.e. the hole lower limit is zero. Earlier systems had bilateral tolerances (e.g. Newall). Unilateral hole tolerances are preferred because, since most fits are clearance fits, the basic deviation (i.e. difference between hole low limit and (clearance) shaft

high limit) is much simpler to identify and determine. If it had happened that most fits were interference fits, a unilateral negative hole (i.e. upper limit is zero) would have been logical. It is convenient with the unilateral system for all holes to have a zero as one limit as it is an additional means of identifying a hole tolerance. Anyway, there is no logical method of determining the lower limit of a hole if it is not zero.

An important fact to bear in mind is that in most manufacturing processes, a male shaft member can be more easily produced than a female hole member (broaching of splines is probably an exception). Thus it is usual to associate a hole with one quality better (or more severe) a shaft. The hardship of the tolerance is thus the same in both cases. This must be borne in mind when setting special tolerances.

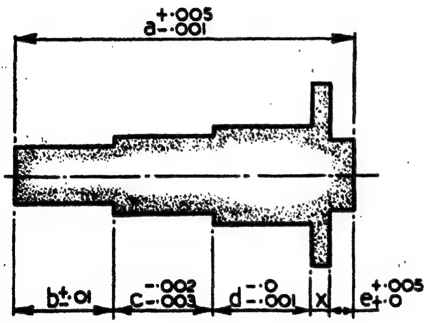


FIG. 34. THE ADDITION AND SUBTRACTION OF LIMITS

As mentioned previously, proper fits are only obtained with certain combination of shaft and hole; this is particularly so with press fits and full use of the information given with the selection and tolerance tables should be made.

4. Dimensioning of Drawings in Relation to Interchangeability

Although the correct practice in the dimensioning of drawings is a subject for study on its own, it has a great effect on tolerance problems, particularly as regards interchangeability, and a few remarks on good and bad practice will not be out of place. It is quite possible in a succession of dimensions which must be added together to give, for example, an overall length to have a final error of several times the general tolerance, often as much as $\pm$ 0·15 in. ($\pm$ 4 mm). Very careful choice of reference faces and the avoidance of additive dimensions is needed to avoid such difficulties on complicated components such as crankshafts.

At this stage it should be pointed out that many engineers do not realize that limits can be added and subtracted to get an answer similarly expressed, without having to add the limits to the nominal dimension in the first place; this should be obvious by elementary mathematics but is surprisingly often overlooked. The following example will help to illustrate this (Fig. 34).

Problem. Determine tolerance on flange thickness x.

$$\text{Nominal thickness} = x = a - (b + c + d + e)$$

Add tolerances on b, c, d, and e—

$$+0{\cdot}01, \quad \begin{matrix}-0{\cdot}002, \\ -0{\cdot}003,\end{matrix} \quad \begin{matrix}-0 \\ -0{\cdot}001\end{matrix} \quad \begin{matrix}+0{\cdot}005 \\ +0\end{matrix}$$

Add all high limits $= +0{\cdot}01, -0{\cdot}002, -0, +0{\cdot}005 = +0{\cdot}013$

Add all low limits $= -0{\cdot}01, -0{\cdot}003, -0{\cdot}001, +0 = -0{\cdot}014$

Tolerance on x is therefore $\begin{matrix}(+0{\cdot}005) \\ (-0{\cdot}001)\end{matrix} - \begin{matrix}(+0{\cdot}013) \\ (-0{\cdot}014)\end{matrix}$

The *low* limit must be subtracted from the *high*, due regard being paid to the algebraic sign.

Thus high limit of $x = +0{\cdot}005 - (-0{\cdot}014) = +0{\cdot}019$

low limit of $x = -0{\cdot}001 - (+0{\cdot}013) = -0{\cdot}014$

Tolerance on x is therefore $\begin{matrix}+0{\cdot}019 \\ -0{\cdot}014\end{matrix}$

Complete understanding of this simple process is necessary on setting any limits for interchangeability.

Notes on Dimensioning

A dimension should only be given once on a drawing and on that view which defines the region being dimensioned the clearest.

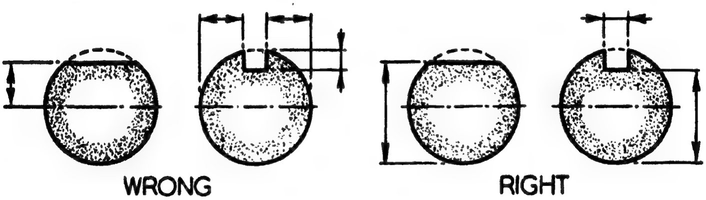


Fig. 35. The Right and Wrong Methods of Dimensioning

Only repeat a dimension if this is vital to the comprehension of the drawing (this is rare). A repeated dimension is invariably forgotten when modifications to the dimension or its tolerance are made.

It is not sufficient merely to put down dimensions in a haphazard fashion to suit the draughtsman, even on a part where clearances are large and fouling due to additive tolerances is unlikely. *The fundamental principle is that a dimension with its tolerance should be so placed that it can most easily be achieved in manufacture.* A common mistake is to put dimensions on a drawing in such a manner that gauging is simplified, but, except in special cases, this is wrong; the special cases are where a machine is set up so that the work coming off it fits or passes a particular gauge, and not a normal dimension.

It can be said therefore that, in general, a dimension should be such as can be measured directly on the part. Fig. 35 shows the right and wrong methods.

In considering interchangeability of length dimensions, the first step is to set down on the drawing those main dimensions which affect interchangeability working off location or reference faces, or those dimensions which are of major importance for performance reasons. Then add in secondary dimensions which are of major importance for performance reasons. Then add in secondary

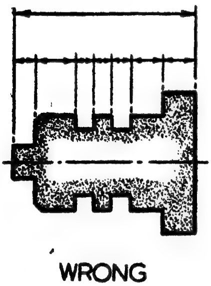


Fig. 36

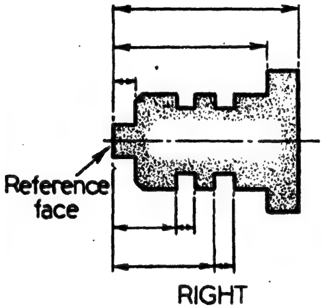


Fig. 37

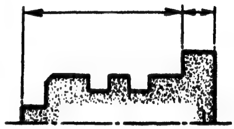


Fig. 38

dimensions with due regard to the possibility of contradictory or additive dimensions.

Example 1

The part shown in Fig. 36 is incorrectly dimensioned for the following reasons—

(i) Contradictory additive dimensions.
(ii) Too many tolerances can be added to position the grooves, possibly affecting interchangeability.
(iii) Position of grooves difficult to determine by machine operator.

Fig. 37 shows the correct method of dimensioning this part. It will be seen that the tool for producing the grooves can be set from the end reference face.

It may, in some cases, be easier for the operator if the overall length is omitted and the flange dimensioned as in Fig. 38. This is because a second operation may be made in facing the end, using the inner face of the flange as a second reference face.

In general, however, this method should not be adopted without special reasons, and only when the draughtsman is sure of the manufacturing process.

Example 2

Fig. 39 shows a bolt for which the dimensions are numbered in order of manufacture, as follows—

Face to give first reference face.

1. Turn bolt diameter.

2. Position undercut of width 3 so that dimension 2 is controlled. (*Note:* if (2) is unimportant, dimension to beginning of undercut, as this simplifies setting of undercut tool.) (See scrap view.)

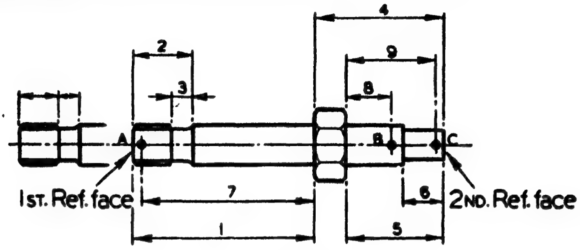


FIG. 39

4. Part off and face to this length to give second reference face.

5. Turn larger diameter full length.

6. Turn small diameter.

7. Drill hole using bolt head as reference face, since this will be used for jig location.

8. Drill hole using the other side of the bolt head as a reference face.

9. In the case of hole *C*, if its position is unimportant, dimension as shown, since this will be done in the same jig as hole *B*. From the jig point of view, the distance between the holes is required, but this is not

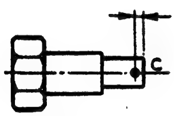


FIG. 40

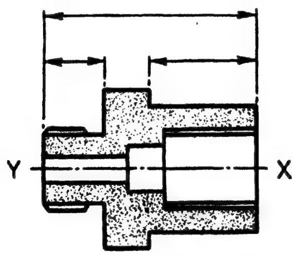


FIG. 41

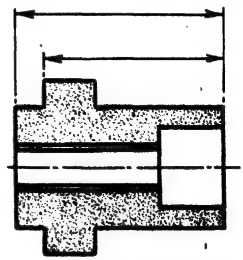


FIG. 42

put on the drawing as it is difficult to measure on the part. The dimension (9) is the best from the shop's point of view. If the position of this hole is important, dimensioning from a third reference face, as the case may be, will be necessary (see Fig. 40).

EXAMPLE 3

In the case of hollow parts the position is not so straightforward, and the method of manufacture is not always evident. For example Fig. 41.

This adaptor would probably be made from the direction right to left and the end face "*X*" would probably be the reference face.

The screw thread on the left-hand end would be done as a second

operation from the face "Y," leaving the width of the hexagon in the middle undimensioned. On the other hand a similar part, as in Fig. 42, would probably be done from the same direction because of the outside diameter step and the counterbore. The small spigot on the left could be made just before parting off and should therefore be dimensioned as shown.

In cases like these the works should be consulted.

EXAMPLE 4

This is a case (Fig. 43) where the designer in conjunction and by agreement with the production engineer, should specify a difficult dimension

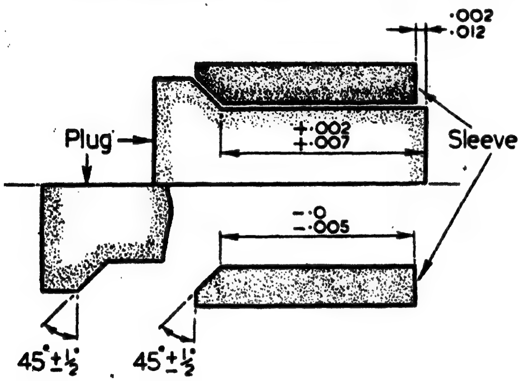


FIG. 43

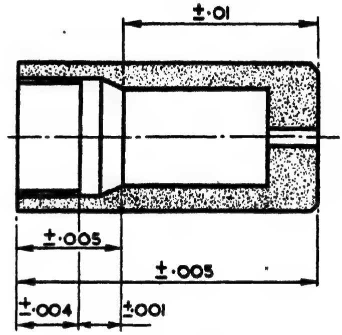


FIG. 44

to measure in order more easily to attain his object, the production engineer taking care of the manufacturing aspect.

The plug member fits inside a sleeve on accurately machined chamfers. The projection of the plug from the other end of the sleeve is important and must be controlled between 0·002 in. and 0·012 in. The plug is dimensioned $\begin{pmatrix}+ 0{\cdot}007\\ + 0{\cdot}002\end{pmatrix}$ as shown from a rather vague point where the diameter runs into the cone. The sleeve is dimensioned $\begin{pmatrix}- 0\\ - 0{\cdot}005\end{pmatrix}$ in a similar manner.

Although both dimensions are almost impossible for direct measurement, the production process involves putting each in a lathe fixture with a length location on the cone, and facing to length, using a setting gauge off the fixture. Thus the dimensions given (angles and lengths) are relevant because they enable the jig to be designed and made. Final gauging is done in a similar manner off the cones or chamfers.

Strictly speaking, in this example angle errors in one direction cause the chamfers to contact on the outer edges and reduce the projections of the plug; the effect is small, however, and in practice is ignored.

EXAMPLE 5

In this case (Fig. 44) the position of the inner end of the internal chamfer in relation to the right hand end of the component is important, because another part fits inside. The designer might think that this dimension should be given as the important one, thinking (quite rightly),

that any other method of dimensioning would involve closer tolerances to get the same final result. However, to achieve the result he wants, a special jig (and gauge) would be needed, introducing an operation of facing to length off the chamfer or cone as a location (ignoring angle errors). Rather than do this, the production man would prefer half the tolerance on two other dimensions readily obtainable. The new dimension to the end of the chamfer would be difficult to measure directly but would be produced with the form tool used to undercut the thread, and thus what would really be used would be three dimensions as shown.

EQUAL DIMENSIONS. This sort of dimension is bad and not sufficiently precise—

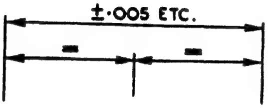


FIG. 45

For general work where the equality of the dimensions is not very important, this is better—

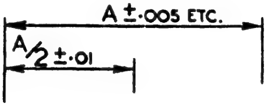


FIG. 46

For greater accuracy, do this—

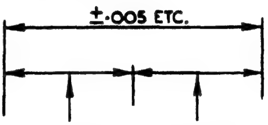


FIG. 47

COUNTERSUNK HOLES. These should be specified as in Fig. 48.

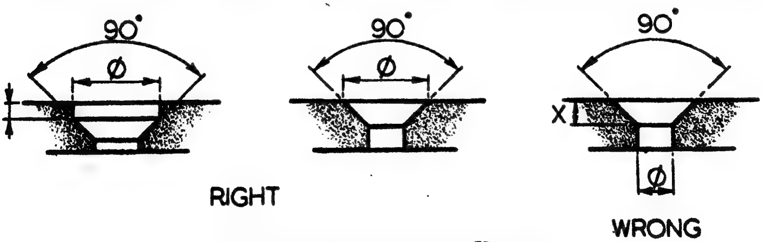


FIG. 48. COUNTERSUNK HOLES

MATING BOLT HOLES. Since holes must be jig drilled to allow two mating parts to be assembled without reaming, etc., the accuracy

of pitching must be specified to the required amount, taking into account any clearances. The correct wording on the drawing should be as follows, the dimensions themselves (p.c.d., etc.) being untoleranced—

"Holes to be correctly pitched within 0·001 (etc.) of their nominal position."

(*Note.* Not $\pm$ 0·001.)

5. Geometrical or Macro Errors

The problem of manufacturing accuracy, not in relation to the size of a particular section, but as regards this section in relation to others, is one of the worst both for manufacture and the designer. These are sometimes known as macro errors as opposed to micro or surface finish errors.

Macro errors of normal parts can be classified as follows—

Errors of *circularity* of a section (ovality is a particular case): non-circular sections have equivalent defects.

Errors of *flatness* or *straightness* of a surface, with the special case of parallelism.

Errors of *axial truth* of a cylindrical (or equivalent) hole or shaft, particular cases being taper, bow or barrel.

Errors of *concentricity* of two or more sections relative to one another (or strictly *co-axiality*).

It is very difficult to make a recommendation to the designer as to what should be done in the way of limiting these errors on drawings. It is common drawing office practice in Britain not to specify errors of concentricity or parallelism, but to leave this to the discretion of the production engineer and inspector who, it is often wrongly assumed, will attend to the matter. It is often claimed, particularly in U.S.A., that this is wrong and that the drawing should be complete and unambiguous in every detail. How often in this country has a sub-contractor working from some other firm's drawings had to have the services of an engineer from the parent company to interpret the drawings, and provide additional information not to be found on the drawings.

There can be no contesting that a drawing should, in principle, be complete, but the difficulty is actually to make it so. The very highest degree of skill and knowledge of the manufacturing process is required by the designer making the drawing and setting the limits, and since the average factory has not the necessary skilled personnel available, little or nothing can be done. Certainly it is better not to attempt elaboration of limits than to do it incorrectly,

as the works will only be involved in a great deal of unnecessary processes and inspection.

The alternative to adequate tolerancing is to rely on the production engineer adopting (possibly after many a mistake) a manufacturing process which automatically achieves the necessary standard of truth as regards flatness, concentricity, or parallelism, etc.

The following diagrams illustrate three drawings of a bolt: (1) typical of a tool maker (Fig. 49); (2) with every conceivable

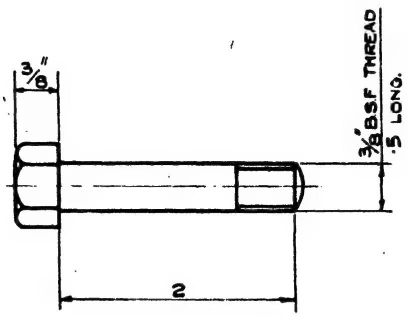


FIG. 49. DIMENSIONING—A BOLT AS DRAWN FOR A TOOLMAKER

error limited (Fig. 50); and (3) what could be regarded as a reasonable compromise, but not, let it be noted, by any means "complete" (Fig. 51).

As usual in such problems a compromise is the most satisfactory procedure to adopt; where the designer wants certain aspects watched specially he should say so, particularly as regards concentricity, but he should not add notes to drawings without the agreement of the production engineer.

In the paragraphs that follow, some of the main errors are discussed.

(*a*) *Errors of Circularity.*

Limits to errors in circularity are implicit in normal limit systems as already explained on page 5.

(*b*) *Errors of Flatness.*

Errors of flatness itself seldom concern the normal designer, although the jig designer meets them frequently. A well-known example is a machine slide where the bed is ground or scraped flat. The lapped surface of slip blocks is a very special case. Flatness of crankcase or cylinder head surfaces on engines necessary to maintain an oil or gas seal is again common.

Errors of parallelism are perhaps more often encountered—the side faces of cylindrical components such as gears, rollers, and the like; or the parallelism of a cylinder block on a multi-cylinder engine.

Errors of this type are best specified as a maximum error in

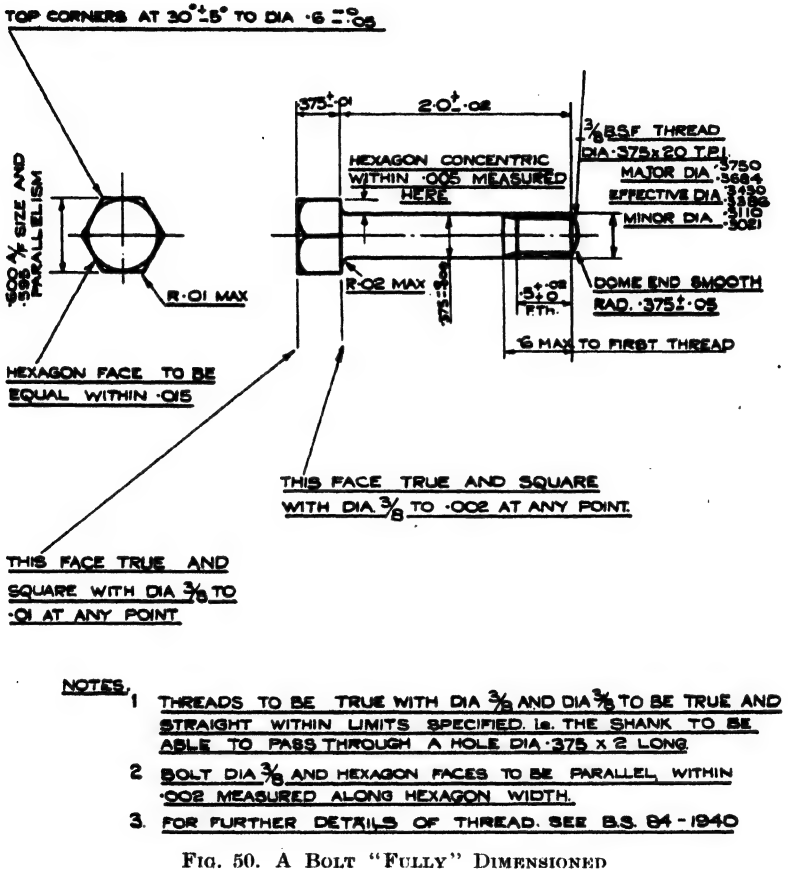


FIG. 50. A BOLT "FULLY" DIMENSIONED

units of 0·001 in. Dial gauges are commonly used with the component on a surface table where convenient. In the case of the bolt head illustrated in Fig. 50 above, a gauge would be used into which the bolt is dropped and the truth of the underside of the head checked optically or with feelers. In the case of a parallel slab, the variation of thickness would show one error, but not the presence of curvature, which would require checking with a dial gauge on a

surface table. In this case a simple check for flatness with a dial gauge would suffice, since this indicates the spacing of two parallel planes through which the slab will pass and the maximum clearance between one surface and a plane.

(c) *Errors of Axial Truth*

As already explained in Chapter I (page 6), the normal limits applied to a shaft or hole really should take into account *axial errors of truth*—straightness, taper, etc. That is, the upper limit of a shaft should refer to the diameter of a perfect cylinder which will just envelop the shaft; or in the case of a hole, the lower limit

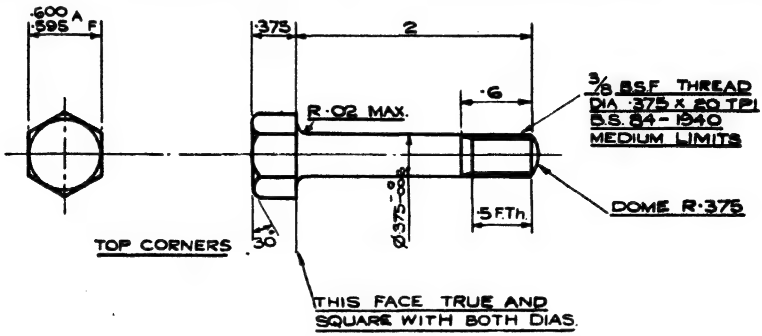


FIG. 51. A BOLT ADEQUATELY DIMENSIONED

should be the diameter of the perfect cylinder (or plug) which will just pass through the hole.

However, modern gauging practice does not take this into account, except in so far as a plug "Go" gauge is usually of reasonable length (1·5–2·0 × diameter) and will be adequate on a hole of the same or lesser length.

Some regard must be paid to the dimensions of the mating member when considering the accuracy of long holes or shafts, since it would be absurd to demand great axial truth on a long cylinder when the piston moving down it was short; it is better therefore to take into account the function of the part and specify an accuracy to be checked by a gauge which is something like the mating part. This type of error is quite common in certain special forms; taper is the most usual due to inaccuracies of the machine (e.g. tailstock malalignment on a lathe) and parallelism of a shaft or hole within 0·001 in. in 12 in. is quite good. Bow or curvature occurs commonly on tubes when machined, due to release of drawing or rolling stresses. A centreless grinding machine may produce curved work. Barrel or double taper distortion is met less frequently

and is usually due to a peculiarity of the component being machined (e.g. lack of rigidity).

(d) *Errors of Concentricity*

These are the most common and generally the most important to limit and control. In many machining operations one diameter is produced at a different set-up from another and it is quite easy to have errors of several thousandths of an inch between the two axes. A simple bush even if the bore and outside diameter are machined at the same chucking, may have wall thickness variation of 0·001 in. Even when precautions are taken to machine the outside diameter of a bush using a mandrel in the bore, the tolerance on the bore itself will mean that the bush may be slightly slack on the mandrel, which must be machined to the lower limit of the hole, and thus the eccentricity of the two diameters will be the tolerance on the bore. For the greatest accuracy and to avoid having several sizes of mandrel, the work is often set up accurately on the mandrel by clocking the component before clamping up. Alternatively a slow taper mandrel can be used.

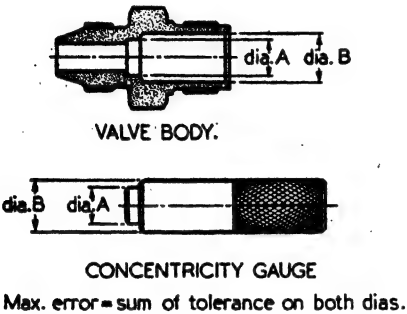


Fig. 52. A Concentricity Gauge for a Valve Body

It should here be made clear how an error of concentricity should be specified. Strictly, concentricity refers to the displacement of the two axes of the diameters involved, which is *half* the error as measured on a dial gauge. It is best to specify all concentricities in terms of dial readings by a note on the drawing of this type: "Diameter *A* and *B* to be concentric to 0·002 in. dial reading."

No trouble should be spared in specifying concentricity accuracy on drawings, as production engineers and inspectors go to a great deal of trouble to control it, and the designer should usually be able to give the shops more latitude than they would normally have taken without guidance.

Although dial gauges are invariably used for checking concentricity of shafts and large holes, many small parts cannot be "clocked" as they are too small to allow the plunger of the gauge to be applied. An example of this is shown in Fig. 52, which is a non-return valve body requiring a true valve seat. For relatively coarse checks of concentricity a double plug gauge as shown could be used, but again the tolerances on the two bores involved restrict

the accuracy of the gauge, unless a very large number of gauges is available, which is absurd. In this case good practice would be to specify on the drawings that the two diameters "must be concentric" without specifying by how much. This note would make the production engineer devise special tools and the inspector see that they were used.

Another typically awkward component to manufacture accurately is the X-shaped spider of a universal joint as shown in Fig. 53. In this case all four journal diameters must be at right angles, co-planar, and the axes must meet at a point. If two opposite pins

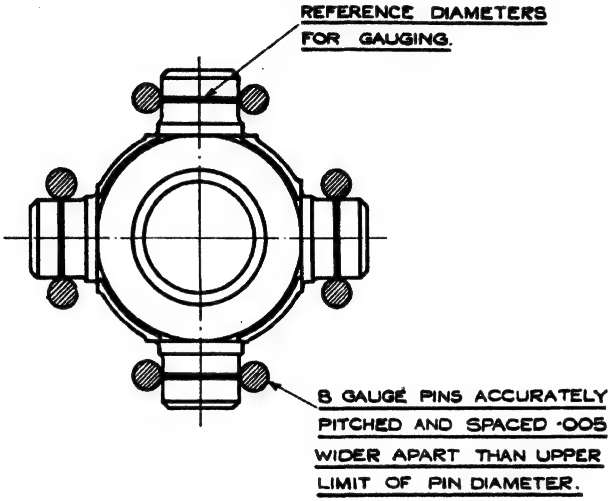


FIG. 53. GAUGING A UNIVERSAL JOINT SPIDER

are "clocked" with the spider in centres, concentricity errors may not be noticed unless the dial gauge is moved along each pin, which is obviously not very practicable. A better method is to rotate the spider with one pin held in a collet or vee-block and to "clock" the other. If this check is satisfactory for all pins, the angles can be checked by, for example, dropping the spider on a plate containing eight pins. The best method of specifying limits on the drawing would be as follows—

> "Opposite pin diameters to be concentric to 0·002 in. (etc.) dial reading, one diameter being rotated true.
>
> "The angles between the four pins to be such that the central reference diameters of the pin cylinders shall not depart from their nominal positions (at maximum diameter) by 0·005 in. (etc.)."

The whole problem of limiting concentricity or truth errors is a difficult one and should be studied in each particular case along the above lines, but in full conjunction with the production engineer and inspector.

6. Selective Assembly Methods

Under this heading are considered components which are made to reasonable machining tolerances and then graded in size subdivisions of these tolerances to obtain greater precision of fit, as well as parts which are adjusted to more precise fit or relation by means of shims and the like.

The extreme case of "shim adjustment" where fully variable adjustment can be made by screws, etc., is not considered, as this eliminates the problem altogether. An example of this is the normal method of screw adjustment on a bevel and pinion to enable the correct gear meshing to be achieved. Another is the typical mounting of Timken taper roller bearings with a screw adjustment of bearing play.

Selective assembly is universal in ball and roller bearing manufacture, since clearances between the balls and the races, accurate to about 0·00005 in. (50 micro-inches) are required and this means tolerances of about 15–20 micro-inches for each of the races and the balls, which is obviously impracticable. On the other hand, if the races and balls are made to normal accurate tolerances (0·0001 in.) and then graded into several sizes within this tolerance, they can be selected to be assembled with the necessary degree of precision. The grading of the balls is relatively easy, one method being to roll them down two metal edges forming a gap, tapering from one end to the other by the tolerance on the ball (say 0·0001 in.); when the ball reaches the position where the gap equals its diameter it falls through into a box or chute. By having 10 boxes in, say, 10-inch length of runner, the balls can be quickly and continuously graded in steps of 0·00001 in. Grading of the races can be by direct measurement or by trial and error on assembly with various size balls.

Selective assembly of motor car engine pistons, piston pins, rings and cylinders is also common as it enables fits accurate to 0·0005 in. to be obtained without the absurdly precise tolerances necessary for full interchangeability. A reasonable tolerance on the cylinder bore would be about 0·001 in., and on the piston diameter 0·0005 in. This will give a variation of fit of 0·0015 in., which may be three times more than is required.

To understand more precisely how selective assembly limits can be determined to improve this state of affairs, refer to Fig. 54. It will be clear from this diagram that the tolerance on the fit required

is the *sum* of the tolerances on the shaft and hole graduations. Assuming that the 0·0005 in. in an example is to be divided equally between shaft and hole, this means that the tolerance on the shaft and hole graduations is 0·00025 in. and that four hole graduations are thus necessary for a work tolerance of 0·001 in. If, as is illustrated in the diagram, the fit tolerance is not divided equally between hole and shaft and more tolerance is given to the hole graduation to decrease the number of graduations or increase the total tolerance on the hole, then the shaft limits are obviously

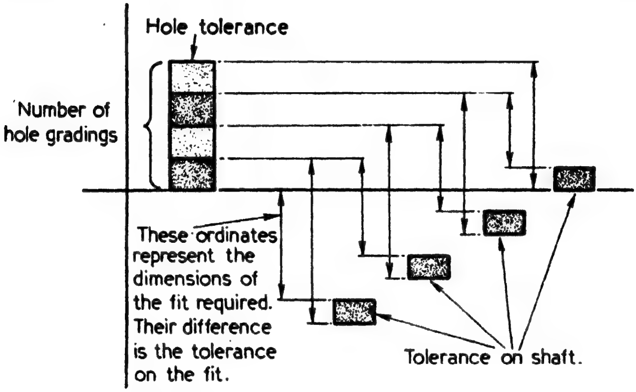


FIG. 54. SELECTIVE ASSEMBLY LIMITS

discontinuous. This is most objectionable, since it means making shafts of several sizes and to very close limits, and in fact cannot be considered as a reasonable process to adopt under any circumstances. The tolerance on shaft and hole must therefore be the same.

It is accordingly possible to formulate the prime fact in connection with predetermined selective assembly limits that—

The number of graduation of the hole (or shaft)

$$= \frac{2 \times \text{the tolerance on the hole (or shaft)}}{\text{accuracy of fit required}}$$

In practice, three graduations is the usual number, and in the case of motor car engine pistons and cylinders the tolerances involved would be approximately 0·0009 in. on the hole and shaft, graded in steps of 0·0003 in. and fitted to an accuracy of 0·0006 in.

The rather paradoxical state of affairs therefore exists that an accurate fit will inevitably involve very accurate workmanship even with selective assembly.

With graded components of the type just discussed it is necessary

to mark them so that the proper mating parts can be fitted together, particularly for service replacement.

Another very common and much simpler process is to provide the assembler with large numbers of components and to allow him to assemble by trial and error; often some crude grading into, say, large and small holes or shafts will help. In this case, the assembler selects a shaft to suit a particular hole and his skill is relied upon to achieve the necessary fit. This process is satisfactory for small parts but too impracticable in the case of the previous example where sufficient pistons could not reasonably be made available.

Selective assembly by trial and error on press fit bushes and on interference fit studs are two common examples of this process.

What may be called semi-selective assembly makes use of reasonable tolerances on mating parts and adjusts the fit with shims or packing washers. The three main applications of shims are on—

Radial adjustment of a journal bearing.
Rotational adjustment or adjustment of orientation.
Linear adjustment.

An example of the first is on engine bearings where between the two parts of a connecting rod or main bearing are shims, which can be removed to allow the bearing shells to be brought closer together. This process is rare nowadays as after removal of shims the bearing has to be scraped in to correct the circularity.

An example of the second application is when two parts screwed together must be screwed home tight and yet the relative orientation must be correct. It is possible to screw parts with the threads starting at the same place on each component (an example is on pipe connection elbow fittings with taper threads which are sometimes, on automobile components, made so that the elbow orients correctly on every assembly even when screwed home tight); it is usually preferred to avoid the problem altogether, or if this is not possible, shims between the abutment faces of the two components will enable the relative orientation to be adjusted. Since one turn equals the pitch of the thread, a nominal shim of, say, one thread pitch in width can be fitted and by adjusting its thickness up to plus or minus half a pitch the correct relation can be achieved. It should be noted that the length of the assembly will vary by an equivalent amount.

The third application to linear adjustment is the most common and is used in all types of engineering product where play, side clearance, backlash, etc., has to be controlled.

Shims are usually made in spring steel sheet (shim or pen steel)

or brass sheet, which are commonly available in the following thicknesses—

0·002, 0·003, 0·004, 0·005, 0·006, 0·008, 0·010, 0·012 in., and then in standard sheet thicknesses.

Sheet 0·001 or 0·0015 in. thick can be obtained but is too thin for normal assembly, and is rarely needed anyway.

Too many thin shims should be avoided and one thick one is better than two thinner ones.

Using combinations of shims almost any size can be obtained, although, as stated above, thick shims should be available to avoid too many shims. Table 5 shows various combinations of shim and what can be obtained.

Another type of shim is the laminated type which has thin foil shims stuck together; when an adjustment has to be made shims are peeled off as required.

PRESS FITS

A press fit has already been defined as a fit where the shaft is always larger than the hole, although the case where the lower limit of the shaft is exactly the upper limit of the hole is also included.

Press fits are used in a variety of applications, perhaps the most common being a bush pressed into a casing or body; in this case the fit is not so critically important as between railway wheel tyres and hubs, the component cylinders of gun barrels, or, perhaps the most critical of all, as on built-up crankshafts for oil or marine engines, where the crank pins, journals and webs are assembled without keying and the grip due to the interference fit is used to transmit the whole torque.

In most cases the actual interference cannot be allowed to vary much as the grip itself must be held between close limits—excessive interference causing permanent distortion of the mating faces; insufficient interference conversely giving insufficient grip. It is a fact, therefore, that press fits demand close tolerances and a higher standard of workmanship than on most other classes of fit. It will also be apparent that proper press fits can only be obtained with certain specified holes and shafts on any tolerance system, and the standard combinations should not be departed from.

1. Determination of Basic Deviation

Since, other factors being equal, the grip resulting from a press fit is substantially proportional to the diameter, some tolerance systems have determined the basic deviation, that is, the minimum interference, on this basis. The smallest practicable variation of tolerance is desirable and systems where the tolerance is proportional

TABLE 5

NUMBER OF SHIMS NEEDED TO GIVE A PARTICULAR THICKNESS

Sizes of Shim—Unit 0·001 in.	Thickness—Unit 0·001 in.																												
	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
2, 3, 10, 20	1	1	2	2	2	3	3	3	1	4	2	2	3	3	3	4	4	4	1	5	2	2	3	3	3	4	4	4	2
3, 4, 10, 20	–	1	1	–	2	2	2	3	1	3	3	2	2	4	3	3	3	4	1	4	4	2	2	5	3	3	3	4	2
4, 6, 10, 20	–	–	1	–	1	–	2	–	1	–	3	–	2	–	2	–	3	–	1	–	3	–	2	–	2	–	3	–	2
3, 6, 15, 30	–	1	–	–	1	–	–	2	–	–	2	–	–	1	–	–	2	–	–	2	–	–	3	–	–	3	–	–	1
4, 8, 20	–	–	1	–	–	–	1	–	–	–	2	–	–	–	2	–	–	–	1	–	–	–	2	–	–	–	2	–	–

to the cube root of the diameter are better in this respect than those using the square root.

The first step in selecting a press fit is to determine the *basic deviation*. For unimportant fits in rigid materials such as a bush in a casing or body, it is often sufficient to take the first classified fit, i.e. that where the minimum shaft is only just larger than the maximum hole (e.g., I.S.A. "p6" or B.S.I. "E"); these fits can generally be used without danger of overstressing normal materials (but not some plastics).

For unimportant fits in elastic materials (e.g. in aluminium alloys), rather more interference will be required and the only way of determining them is to make up trial shafts machined exactly to the least interference of the fit under consideration, and determine the suitability of the fit by test. (There is obviously no need to make special holes on top limit, as shafts corresponding to the actual holes available plus the minimum interference can be made up.)

The method of calculating interferences for important fits is rather involved and depends on some factors that cannot be taken into account accurately, such as coefficients of friction, surface finish, etc. The method of calculating the stresses in the components of a press fit has been thoroughly investigated and a very good account of some particular cases is contained in Morley's *Strength of Materials*. These methods are accurate, but the primary result of interest to the designer setting the limits—the slip load—can only be assessed as the theoretical minimum. Once the minimum interference has been determined, the designer must check by calculation that the extreme interference, tolerances included, does not overstress the component members.

If the components are of different materials, the relative Young's moduli and Poisson's ratios have to be considered. The general case of two rings pressed or shrunk together gives the following formula determining the radial pressure between the two mating surfaces—

$$\frac{p}{E'}\left(\frac{R_3^2 + R_1^2}{R_3^2 - R_1^2} + \frac{1}{m'}\right) + \frac{p}{E}\left(\frac{R_1^2 + R_2^2}{R_1^2 - R_2^2} - \frac{1}{m}\right) = \frac{\delta}{2R_1}$$

where p = radial pressure
δ = interference
R_1 = nominal mating radius
R_2 = inner radius of inner ring
R_3 = outer radius of outer ring
E = Young's modulus, inner tube
E' = Young's modulus, outer tube
$1/m$ = Poisson's ratio, inner tube
$1/m'$ = Poisson's ratio, outer tube

If the materials are the same, the formula becomes

$$\frac{p}{E}\left(\frac{R_3^2 + R_1^2}{R_3^2 - R_1^2} + \frac{R_1^2 + R_2^2}{R_1^2 - R_2^2}\right) = \frac{\delta}{2R_1}$$

If the inner "ring" is a solid shaft, $R_2 = 0$, and

$$\frac{p}{E}\left(\frac{R_3^2 + R_1^2}{R_3^2 - R_1^2} + 1\right) = \frac{\delta}{2R_1}$$

If the outer "ring" has a very large outer radius, R_3, in comparison with the inner radius, R_1, and can be considered as rigid,

$$\frac{p}{E}\left(1 + \frac{R_1^2 + R_2^2}{R_1^2 - R_2^2}\right) = \frac{\delta}{2R_1}$$

Since for all normal purposes the variations in Poisson's ratio can be ignored (if for no other reason than that precise information for various materials is very hard to obtain), the most useful formulae are—

(1) Case of bush pressed into stiff outer housing—

$$p\left[\frac{1\cdot 3}{E'} + \frac{1}{E}\left(\frac{R_1^2 + R_2^2}{R_1^2 - R_2^2} - \frac{3}{10}\right)\right] = \frac{\delta}{2R_1}$$

(2) Case of solid shaft pressed into ring—

$$p\left[\frac{1}{E'}\left(\frac{R_3^2 + R_1^2}{R_3^2 - R_1^2} + \frac{3}{10}\right) + \frac{7}{10E}\right] = \frac{\delta}{2R_1}$$

EXAMPLE

A solid duralumin shaft 2 in. diameter is pressed into a steel ring of outer diameter 4 in. The interference is 0·002 in. Find the radial pressure [$E' = 30 \times 10^6$ lb./sq. in.; $E = 10 \times 10^6$ lb./sq. in.].

$$p\left[\frac{1}{30 \,.\, 10^6}\left(\frac{4+1}{4-1} + \frac{1}{4}\right) + \frac{3}{4 \,.\, 10 \,.\, 10^6}\right] = \frac{0\cdot 002}{2}$$

$$p = \frac{0\cdot 002}{2} \times \frac{900}{122} \times 10^6 = 7400 \text{ lb./sq. in.}$$

In the case of components subjected to heat or cold, working stress (as internal pressure in a gun) or centrifugal forces (as in a railway tyre), additional allowances will have to be made for thermal or working stresses which affect the radial pressure. The above formulae cannot be applied with any precision to determine the exact grip, but they do enable an approximate estimate of the stresses set up by an interference fit to be made; it is essential for proper assembly to keep these stresses within the elastic range of the material as otherwise the actual radial pressure will be reduced by one member deforming permanently. Excessive interference is thus to be avoided.

2. Factors Affecting Grip

From the above formulae it is obvious that the two fundamental factors which affect the grip between two press-fit components are the elastic limit or proof strength of the material, and the magnitude of the modulus of elasticity or Young's modulus. Many materials have unusually low elastic limits (as a percentage of ultimate strength)—an example being aluminium bronze—and this will limit the permissible radial pressure as compared with another material of similar properties except that its elastic limit is greater.

The interference required to give a specified radial pressure is inversely proportional to the Young's modulus. It will be seen for example from the table in Appendix 3, which lists the moduli of several common materials, that an interference three times greater is required for a bush pressed into duralumin as compared with steel, to achieve a given radial pressure. In practice heavy fits are not usual on materials with higher elasticity, and it is good practice to use a shaft one or two fits larger in these cases than for steel.

The grip due to a given radial pressure will be proportional to the coefficient of friction. There is some evidence that abnormally high coefficients are sometimes encountered (even greater than unity), indicating overlapping of the machining irregularities of the two surfaces, and it is doubtful if the normally listed coefficients of dry metals in contact can be used except to indicate the minimum grip.

Machining accuracy, particularly in regard to parallelism and truth of the two co-operating surfaces of an interference fit, has a profound effect on the grip for obvious reasons. In the case of an oval shaft and an oval hole, different fits will occur when the ovality corresponds or is 90° out of phase. On the other hand, ovality in a thin bush pressed into a rigid casing is unimportant if its mean diameter is correct.

Particular care must be taken to see that the parts involved are true at least within the drawing limits; for critical fits, particularly on large diameters, it may even be necessary to call for special accuracy as regards parallelism or freedom from taper or barrel.

The surface finish of the two mating parts in many press fits is important, not so much to ensure grip (roughness may even help this in some cases), but to ensure that the fit can be dismantled without dimensional distortion of the surfaces. In the case of ball race housings, for example, every time the bearing is pressed in, the surface of the housing is compressed, and if machined with perceptible roughness the peaks of the irregularities will be "ironed" into the cavities, causing a slight increase in size and a reduction

of interference. A ground or smooth surface will be free from this defect and can be dismantled several times without noticeable effect on the fit.

On press fits, therefore, where dismantling may occur, the best finish practicable should be specified by the designer.

The effect of lubricants used during assembly of interference fit components has been investigated in detail by Russell (*Proc. Inst. Mech. Eng.*, Vol. 129, 1934), and some interesting conclusions derived. When two components are pressed together suitably lubricated, a film of lubricant remains between the surfaces even at the high bearing pressures of a heavy press fit. This lubricant has a profound effect on the grip of the fit and enables the parts to be pressed apart without surface damage, or even marking in some cases. The type of oil used also caused a difference of grip of, for example, as much as 100 per cent as between rape oil and Bayonne oil. Even the water film occurring with freeze fit shafts reduced the grip.

The maximum grip occurred when the surfaces were perfectly clean and dry, a condition obtained in the tests in question by washing with soap and hot water, but probably obtainable by commercial degreasing. Whether the lubricating effect of the oil film remains after a long time remains to be investigated on further tests, although it is not thought so; other effects such as fretting corrosion have been seen in many instances. The best advice that can be given is to determine the fit for a given performance with all variables as closely controlled as possible, and to make use of the dry and clean grip only in those cases, such as engine crankshafts, where dismantling is not necessary, since this may (in fact probably will) cause tearing or seizure of the two surfaces unless local expansion of the outer member by heating can temporarily reduce the interference.

The exact method of assembly (see next paragraph), whether by force, by contraction of the female member after heating, or by expansion of the male member after freezing, has little effect on the resulting grip. Force fits may cause slight "ironing" of the surfaces which would not be obtained during expansion fits, but this effect is negligible when the surface finish is good. As already mentioned, freezing the male member results in a part being covered with a film of frozen moisture which prevents a dry fit.

3. Methods of Assembly

(*a*) *Force Fits*

Force fits are best carried out by means of a hydraulic press as the pressure (and thus the load) can be easily controlled. On the other hand, many mechanics prefer lever or screw presses for delicate

work, as the insertion pressure can be felt and there is less likelihood of damaging the components. Great care has to be taken to keep the components square with each other and properly in line when starting the pressing operation, and the type of insertion arbor illustrated in Fig. 55 (as recommended by self-lubricating bush makers) is of value. In the case of this type of sintered bush the arbor can be used as a means of controlling the bore of the bush after pressing, since the material is substantially compressible.

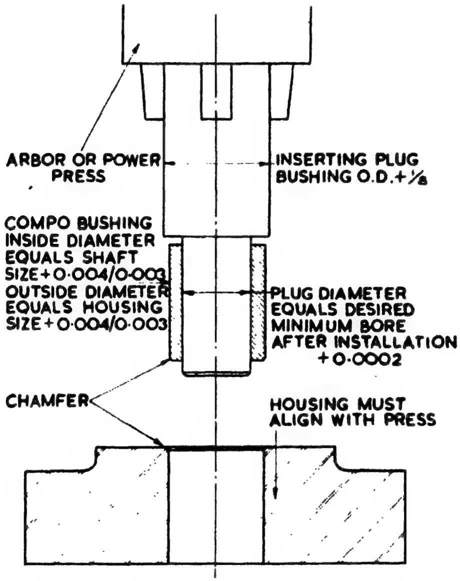


FIG. 55. PILOTED PRESS METHOD FOR INSERTING BUSHES

With a normal bush the arbor should not be too tight on the bore of the bush, otherwise it may have to be pressed out when the bush is in position. It is hardly necessary to stress that on all fits assembled under a press, the whole component should be so supported in the press as to prevent damage and particularly distortion.

In railway crank practice force fits with 10 tons press load per inch of diameter are common. In fact, as high as 15 tons per inch diameter is used in some shops. These correspond to about 0·002 in. per inch diameter interference.

(*b*) *Shrink Fits*

In a shrink fit, the outer female member is heated until it expands sufficiently to make the two parts of the interference fit go together

with little or no effort. A very common example of this is in the assembly of the piston pin in a motor car engine piston, where the differential expansion of aluminium alloy and steel when the piston is put in boiling water enables the pin to be withdrawn easily. The standard method of fitting railway tyres is to heat them on a gas ring and then drop them over the wheel proper and allow them to cool. On a 6 ft. diameter wheel, a temperature rise of 300° C. will give an expansion of nearly $\frac{1}{4}$ in. A similar process is used for gun barrels, gear teeth rings on flywheels, etc., and in fact shrinkage is a very common process in many general engineering shops and presents little problem provided the parts may be heated to 200–300° C.

Hot oil is often used as a convenient means of getting a greater temperature than boiling water, but 150–180° C. should not be exceeded with normal light oils as this approaches the flash-point of the oil, and increases the fire risk.

Appendix 3 lists the coefficients of expansion of normal engineering materials expressed in a convenient manner (units of 0·001 in. per inch per 100° C.). It is convenient to memorize the expansion of steel as equal to the differential expansion of aluminium and steel and both equal to "one thou. per inch per 100° C." The expansion of aluminium itself is twice this.

Due regard must be paid to the effect of temperature on the materials involved. Aluminium or magnesium alloys should not be heated above 200° C., and even this temperature on a large casting may cause distortion.

(c) *Freeze Fits*

Instead of heating the female member, the male member may be contracted by cooling and subsequently allowed to expand into the female. This process uses an industrial refrigerator giving a temperature of about − 50° C., or to obtain lower temperatures the component is cooled in liquid air (boiling point − 190° C.). Examples of this process are in the insertion of exhaust valve seat inserts in engine cylinder heads or blocks, or in the insertion of brass bushes in various assemblies.

The process is convenient only for small parts as otherwise the size of the refrigerator equipment is prohibitive or consumption of liquid air is excessive. However, on suitable parts the process is very convenient, since the temperature is controlled and is unlikely to damage the structure of the material in any way. A combination of freezing and heating is also used as this enables reasonable maximum temperatures to be used.

A convenient method of reaching $-50/-60°$ C. without expensive equipment, and suitable for occasional use, is to cool the part in alcohol to which solid frozen CO_2 ("Drikold") is added.

Appendix 3 can be used, as discussed above, in calculating temperatures for freeze fit assemblies, but it must be remembered that in the very usual case of steel or brass pressed into aluminium, heating the latter is the more effective.

As already mentioned, it is almost impossible to assemble a frozen shaft entirely dry, owing to the frost formation on the component.

4. Assembly Loads with Press Fits

It is not possible to calculate accurately the force required to press on or off a pair of press fit parts, because there are too many variable and unknown factors. However, from the formulae given above it is possible to arrive at an estimate of the radial pressure between the parts (p) for a given interference (δ). The frictional force is thus given by—

$$F = p \,.\, \pi \,.\, D_1 \,.\, b \,.\, \mu$$

where F = frictional or press force, lb.

p = radial pressure, lb./sq. in.

D_1 = nominal mating diameter, in.

b = length of contact, in.

μ = coefficient of friction.

The coefficient of friction on steel to steel fits is about 0·15 on assembly, lubricated, and about 0·2 during disassembly (i.e. requiring 33 per cent more load).

Two common cases can be determined according to the following formulae derived from those on page 64, and using the previous terminology.

(a) Assembling a Ball Bearing on a Steel Shaft

The bearing can be considered as a thin ring whose outer diameter is taken as the mean value, allowing for the ball track.

$$F = \mu \,.\, \frac{\pi \,.\, E}{2} \,.\, \delta \,.\, b\left(1 - \frac{R_1^2}{R_3^2}\right)$$

$$= 7 \,.\, 10^6 \,.\, \delta \,.\, b\left(1 - \frac{R_1^2}{R_3^2}\right)$$

(b) Assembly of a Ball Bearing in a Stiff Housing

Steel housing: $$F = 7\,.\,10^6\,.\,\delta\,.\,b\left(1 - \frac{R_2^2}{R_1^2}\right)$$

Cast iron housing: $$F = 7\,.\,10^6\,.\,\delta\,.\,b\left(\frac{R_1^2 - R_2^2}{3{\cdot}2R_1^2 - 1{\cdot}2R_2^2}\right)$$

Aluminium housing: $$F = 7\,.\,10^6\,.\,\delta\,.\,b\left(\frac{R_1^2 - R_2^2}{4{\cdot}6R_1^2 - 2{\cdot}6R_2^2}\right).$$

Example. A 2 in. diameter bearing ring of mean thickness 0·3 in. and width 0·75 in. is pressed into an aluminium housing with an interference of 0·004 in. Find the press load.

$$F = 7\,.\,10^6\,.\,4\,.\,10^{-3}\,.\,0{\cdot}75\left(\frac{1 - 0{\cdot}7^2}{4{\cdot}6 - 2{\cdot}6\,.\,0{\cdot}7^2}\right)$$

$$= \frac{21\,.\,10^3}{2240\,.\,6{\cdot}5}$$

$$= \underline{\underline{1{\cdot}46 \text{ tons.}}}$$

5. Bore Closure on Bushes

A very common problem in connection with the use of press-fit bushes is the predetermination of the final bore size after pressing in, allowance being made for the closure of the bore. The necessity for final reaming is inconvenient both from the cost point of view and also because it makes service replacement of bushes more laborious.

In the case of self-lubricating sintered bushes, usé can be made of the compressibility of the material to form the bore to size on a mandrel or arbor, as already illustrated in Fig. 55. The mandrel is made 0·0002 in. larger than the final lower limit required on the hole and the material is formed very slightly smaller (about 0·0002 in.) than the arbor, which thus has to be drawn out.

Alternatively, bushes made from rolled strip may be used, and in this case errors of outside or inside diameter are compensated for by variations in the gap between the ends of the bearing. The tolerance on the thickness of the strip is the only other variable introduced, and as this is usually controlled to close limits, the final resultant tolerance can be reasonable.

As an example, if the limits on the housing are $\begin{matrix}+\,0{\cdot}001\\+\,0\end{matrix}$, and on the strip thickness $\pm$ 0·00025, the final hole limits are—

$$\left(\begin{matrix}+\,0{\cdot}001\\+\,0\end{matrix}\right) - 2\left(\pm\,0{\cdot}00025\right) = \left(\begin{matrix}+\,0{\cdot}0015\\-\,0{\cdot}0005\end{matrix}\right)$$

the tolerance being 0·001 + 2(0·0005) = 0·002 in.

It is not possible accurately to predetermine bore closure on ordinary bushes, as it depends on the relative stiffness of the bush and housing, except in the case where the housing is considered as perfectly rigid. A further complication is introduced if the bush's diameters are not concentric, and if the housing or bush diameters are oval. Ignoring this and assuming perfect rigidity of the housing, at least an estimate of the maximum possible closure can be obtained.

The thickness of the bush can be derived by subtracting the two bush diameters, with tolerances, and then deriving the final bore dimensions by subtracting the double thickness from the housing diameter as in the case of the rolled strip bush. This process is certainly inaccurate as it ignores various secondary effects, but it is useful, as mentioned, to indicate the order of the closure.

An example will illustrate the process—

Housing diameter	$A \begin{smallmatrix} +\ 0{\cdot}001 \\ +\ 0 \end{smallmatrix}$ in.
Bush outside diameter	$B \begin{smallmatrix} +\ 0{\cdot}002 \\ +\ 0{\cdot}001 \end{smallmatrix}$ in.
Bush bore	$C \pm 0{\cdot}0005$ in.

The double wall thickness is $B - C$.

The final bore size $A - (B - C)$ is therefore

$$\begin{pmatrix} +\ 0{\cdot}001 \\ +\ 0 \end{pmatrix} - \left\{\begin{pmatrix} +\ 0{\cdot}002 \\ +\ 0{\cdot}001 \end{pmatrix} - \left(\pm\ 0{\cdot}0005\right)\right\}$$

$$= \begin{pmatrix} +\ 0{\cdot}001 \\ +\ 0 \end{pmatrix} - \begin{pmatrix} +\ 0{\cdot}0025 \\ +\ 0{\cdot}0005 \end{pmatrix}$$

$$= \begin{matrix} +\ 0{\cdot}0005 \text{ in.} \\ -\ 0{\cdot}0025 \text{ in.} \end{matrix}$$

In actual practice the greatest change (0·0025 in.) due to the maximum interference will not all be transferred and the probable tolerance would be (+ 0·0005/−0·0015), about 50–75 per cent of the interference being transferred, depending on the relative stiffness of the parts.

The most satisfactory process is to determine the closure by actual trials using parts on extreme limits, but even then other errors such as concentricity, ovality, etc., will upset the result; consequently, ample latitude should be allowed, this meaning very close pre-assembly tolerances, or a coarse final tolerance.

6. Note on Tolerance Tables

In Chapter IV, comprehensive tables are given of all press-fit limits on the common tolerance systems. These range from the

comparatively simple Newall or A.S.A. fits to comprehensive fits of the I.S.A. system. For the average uncritical fit, the fits of least interference can be used without danger, but the heavy interference fits of the I.S.A. system, or those needed to reproduce railway practice, should be used with the greatest care and after careful trial, if possible with the use of strain gauges or, for example, stress indicating paint ("stress-coat"). It is almost impossible to prepare selection tables of heavy press fits, as too many variables have to be taken into account.

TRANSITION FITS

A transition fit has already been defined as one between a press fit and a clearance fit, there being slight clearance or interference depending on the actual size of the two mating parts. It is usual to consider as a transition fit the case where the shaft upper limit is the same as the hole lower limit (usually zero), and therefore although on the average there is clearance, on extreme limits the parts may just be the same size. Most tolerance systems list a few transition fits as these are very useful for a variety of purposes. There is no particular way in which the tolerance on a transition fit should vary with diameter which can be justified on theoretical grounds; variation with the cube root of diameter seems most reasonable. The main applications of this class of fit are listed below.

1. Spigots

A spigot* is a common engineering device, either in the form of a locating diameter, or dowels or pegs, to locate or centre two components together on assembly. Common examples are flange mounted magnetos or starter motors on automobile engines. A clearance fit is adequate for a rough spigot, but for accurate work a transition fit is better, since the maximum clearance is small and the maximum interference not so much that the component cannot be tapped lightly into position. The actual fit selected can be a compromise between clearance and the difficulty in assembling the light press fit.

2. Light Press Fits by Selective Assembly

Where a component has to be assembled with slight interference, a transition fit is often specified and the required fit obtained by random selection. In the transition fits nearest to genuine press fits, the probability is that the parts will assemble with slight interference, and it is not difficult to try, for example, a further

* The English word "spigot" is inadequate to describe a locating diameter; the French word "*centrage*" conveys the meaning better.

shaft if the first one selected is too slack. This process avoids the comparatively heavy interference possible on extreme limits of genuine press fits, although at the expense of complicating assembly and service replacement.

3. Assembly with Least Play

In many cases large clearances are to be avoided and the expense and complication of a very accurate fine clearance fit makes the designer specify a transition fit of reasonable tolerance, the possibility of having lightly to force the parts together being accepted. A connecting rod big end bolt is an example of this. Certain ball race fits are others.

4. Assembly with Least Play Compatible with Ease of Assembly

In this case the "size and size" fit is used, i.e. the fit where the clearance varies from zero and in the case of unilateral hole systems, the shaft upper limit is zero. The clearance is controlled entirely by the magnitude of hole and shaft tolerances and the fit can always be assembled without difficulty. This class of fit is commonly used for lengths of two mating parts (as a bush and housing) where the two parts should be as near as possible of the same length but should not overlap or underlap, as the case may be.

CLEARANCE FITS

A clearance fit is essentially a running fit and the clearance is necessary to permit the two parts to operate together with a film of lubricant between them. The very common clevis pin of a forked shackle joint is an extreme example of this, since the pin is machined with a clearance fit on the (optimistic) assumption that the joint will be lubricated; otherwise a lower transition fit would be used.

It is not the concern of the present volume to discuss the methods of determining the minimum clearance of a running fit, since this is a question of the type of machine, the lubrication system, speeds, loads, etc. Some mention of the problem will be made in general terms, and in particular, the importance of the variation in clearance due to the tolerance. The question of machining finish as opposed to accuracy is also outside the scope of this book, although it can be said that modern practice is to use the very best possible surface finishes for high duty bearings.

1. Variation of Clearance with Diameter

Since any good tolerance system should provide a range of clearance fits to meet all requirements, the basic deviations (i.e. minimum clearances) should be calculated according to varying

rules for different diameters. If they are all proportional to the square root of diameter, the first fit will have considerable minimum clearance when the diameter is large. In the B.S.I. system all basic deviations vary as $D^{\cdot 445}$. In the I.S.A. system the deviations vary as $D^{\cdot 34}$, in the first fit, through various factors in the intermediate fits, to the final ones which vary as the diameter itself. This appears more reasonable since, although with close running fits the clearance should vary little, on slack fits it may vary as the diameter so that the friction of the bearing is at a minimum; information on this subject is somewhat contradictory, however, and it may be impossible to do more than compromise.

2. Minimum Running Clearance

It used to be thought that a bearing required an appreciable running clearance to allow a reasonable oil film thickness. There are one or two modern examples of precision devices with remarkably close fits, which run satisfactorily with oil films of almost molecular thickness. The Diesel injector pump is an example, the plunger and body being lapped or honed to a running clearance of a few microns or about 0·0001 in. Another example is on high pressure hydraulic pumps where pistons and cylinder block rotors operate at similar or even smaller clearances. In fact, the type of fit might be called a "tight plug gauge fit." In these examples the close fits are associated with extremely fine surface finish (1 or 2 micro-inches R.M.S. value).

3. Normal Running Clearance

Normal lubricated bearings, however, use fits of appreciable clearance, particularly as on engine crankshafts, where distortion may be present; these clearances are of the order of 0·001–0·003 in. for small diameters. In the case of pressure lubricated bearings where the lubrication is hydrodynamic, an appreciable clearance enables the oil film to be maintained dynamically and the shaft is deliberately allowed to run slightly eccentrically to assist in its maintenance.

With electric motors, rather closer fits are used, since the relationship of rotor and stator is important to maintain electrical balance.

Fig. 56 shows the maker's recommended clearances (but no tolerances) for self-lubricating bearings.

Fig. 57 is a reproduction of some information given in a paper* by Prof. H. W. Swift, and is of particular interest in that it takes into account various other published information and endeavours to arrive at a satisfactory compromise between the variations in

* *Proc. Inst. Mech. Eng.*, Vol. 129, page 399.

this other information. This curve is strictly correct only for a bearing of a diameter not greater than the calculated optimum and of width 1·5D, but can be used as a guide in other cases.

4. The Effect of Tolerance on Bearing Performance

According to Swift in the paper already mentioned, appreciable variation of the theoretical clearance is not very harmful. To quote

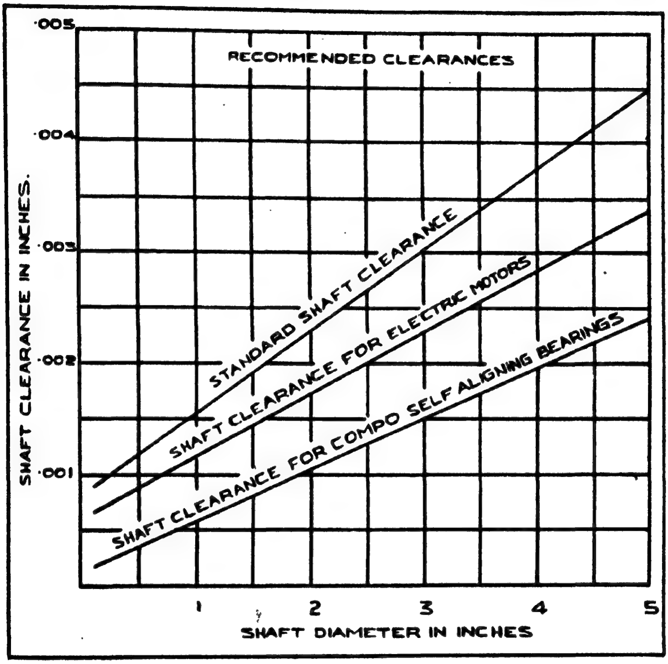


FIG. 56. RECOMMENDED CLEARANCES FOR USE WITH COMPO SINTERED BEARINGS

him, "while errors of ± 20 per cent in the clearance do not affect the friction and the film thickness by more than about 5 per cent, errors of ± 30 per cent involve changes to the extent of 12 per cent, errors of ± 40 per cent changes up to 20 per cent, and errors of ± 50 per cent changes up to 30 per cent. It is therefore clear that although a small percentage error in the clearance has little effect on the performance of a bearing, clearances in which tolerances or errors are likely to exceed about 30 per cent are bound to deprive any scientific design of much of its virtue." It is interesting to note that a reduction in clearance has about the same effect as an increase, but in normal limit determination the tolerances on shaft and hole

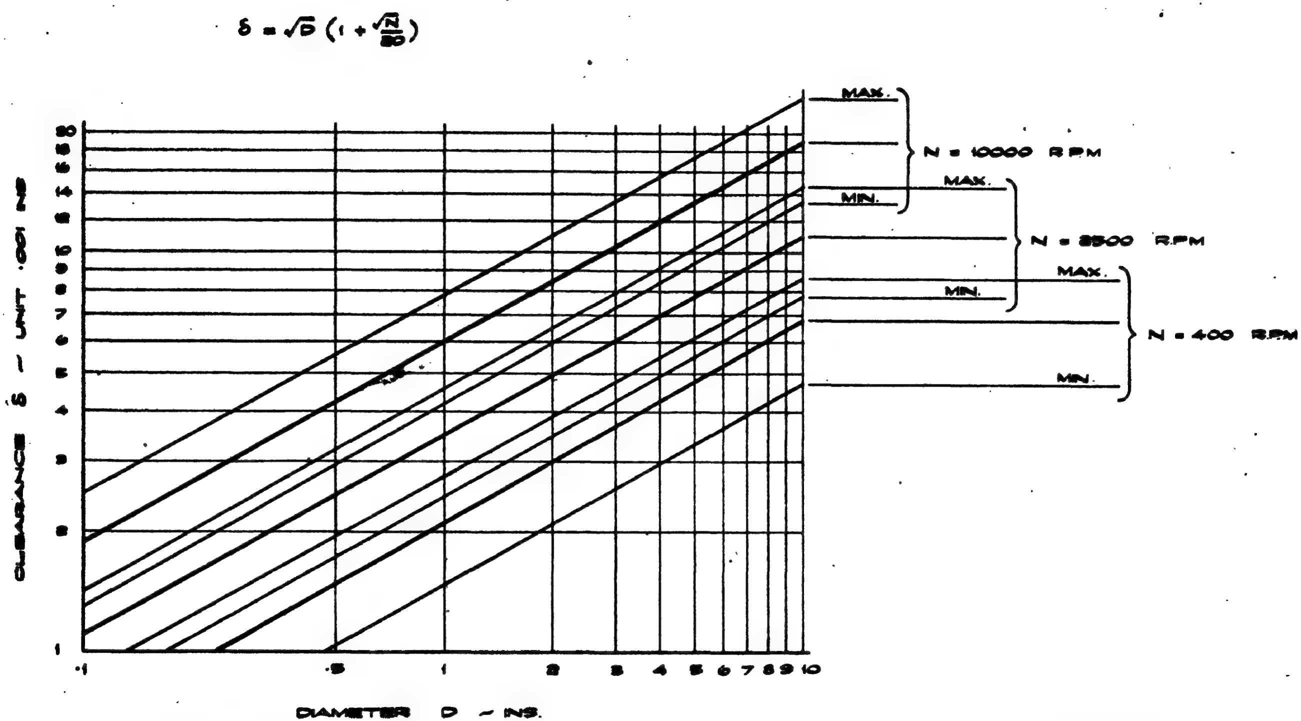


Fig. 57. Prof. H. W. Swift's Recommended Clearances for Properly Lubricated Bearings

increase the clearance over and above the basic deviation. Assuming that the shaft has a closer tolerance than the hole, the results of Fig. 57 correspond to the I.S.A. qualities of tolerance shown in Table 6, as expressed graphically in Fig. 32 (page 31).

TABLE 6

SHAFT TOLERANCE QUALITY

Shaft Diam.	R.P.M. 500	R.P.M. 1000	R.P.M. 3000
in.			
1·0	6	6	6
2·0	6	6	6
4·0	6	6	7

Quality 6 shafts can only be expected to be obtained by fine grinding, but this process is usually specified anyway as a means of ensuring an adequate standard of surface finish.

OVERSIZE AND UNDERSIZE HOLES

1. Oversize Holes

Tables of "oversize" holes (i.e. holes rather larger than the standard unilateral hole with both limits positive) are often given in standard tolerance systems, and in fact a complete range of negative to positive holes corresponding to the normal range of shafts is listed in the I.S.A. system.

2. Application

Oversize holes have two particular applications. In the case where a standard unilateral hole may have worn in service it may be opened up to a standard oversize hole and mated with a corresponding shaft. There is little point in this process as the better and more scientific process would be to increase the nominal size. This is in fact what is normally done, the nominal size being increased by 0·005, 0·010, 0·020, or 0·030 in. for example. Standard shaft limits are then retained with the nominal shaft size increased accordingly.

3. An Important Example

The most important application of an oversize hole is in the case where one shaft is mated with two holes of different fit. As an example of this, a shaft may have two collars on it, one sliding and the other pressed on. Either both holes can be standard and the shaft stepped, or the shaft can be parallel and the holes of

different size. It often occurs that stepping the shaft is more inconvenient than using an oversize hole, and thus the tabulation of this type of hole in a tolerance system is very convenient.

Owing to the large variety of oversize hole and standard shaft combinations it is not practicable to tabulate satisfactorily the cross-referencing of the various fits, and the proper procedure is to

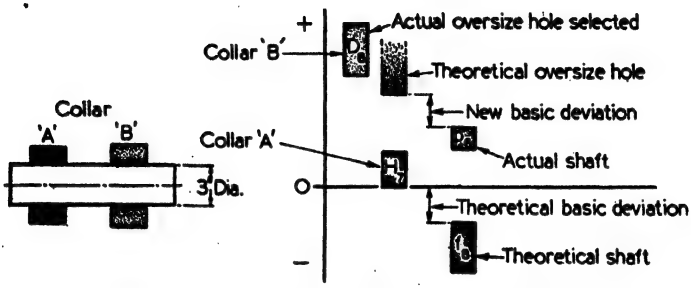


FIG. 58. AN EXAMPLE OF THE USE OF "OVERSIZE" HOLES

determine the oversize hole, by considering the basic deviation of the fit required and seeing which hole gives the closest result.

An example using I.S.A. tolerances will make this more clear (see Fig. 58).

A shaft 3 in. dia. has a press fit collar *A* one end and a sliding collar *B* on the other.

Collar *A*—dia. 3 in. H7	$= 3{\cdot}0 \begin{smallmatrix} +\ 0{\cdot}0012 \\ +\ 0 \end{smallmatrix}$
Shaft size to give press fit—3 in. p6	$= 3{\cdot}0 \begin{smallmatrix} +\ 0{\cdot}002 \\ +\ 0{\cdot}0013 \end{smallmatrix}$
Normal shaft size to give sliding fit—3 in. f8	$= 3{\cdot}0 \begin{smallmatrix} -\ 0{\cdot}0012 \\ -\ 0{\cdot}0030 \end{smallmatrix}$
Basic deviation of clearance fit	$= 0{\cdot}0012$
Therefore lower limit of oversize hole	$= 0{\cdot}002 + 0{\cdot}0012$
	$= 0{\cdot}0032$

The nearest corresponding oversize hole for collar *B* is "D," and this has a lower limit = 0·0039 in.

Although this gives a slightly greater basic deviation, since the tolerance on the press fit shaft (Quality 6) is less than on the normal running fit (Quality 8), Quality 8 applied to the oversize hole will give a maximum clearance about the same as with the clearance shaft and standard hole. The selected oversize hole is then D8, and the maximum clearance for the combination D8/p6 = 0·0044 in., as compared with a theoretical 0·0042 in. for the combination H7/f8.

4. Undersize Holes

Although the common case mentioned above uses an oversize hole, it is conceivable that it might be more convenient to make one hole undersize than the other oversize. For example, if a shaft carries several collars, one only of which needed an interference fit, the others being in the majority might be left standard and the press fit obtained by using an undersize hole. The procedure outlined in the previous paragraph can be adopted.

TOLERANCES ON COMPONENTS INVOLVING ANGLES

As already stated in Chapter I (page 23), angles are very difficult to measure and therefore difficult to tolerance. Perhaps the most common engineering example of an accurate taper is the Morse taper shank on a drill or collet, but these are ground to "blue" an accurate gauge and are not toleranced in the normal way.

Appendix 4 may be useful when dealing with these matters.

1. Tapers

The average experienced engineer can tell from a "blueing" test whether a pair of tapers will be satisfactory, but few will be able to quote the corresponding limits to two reference diameters on the cones. The designer should, however, specify the taper in terms of two diameters a certain distance apart. He should then give limits to one of the diameters (but not to the length) to enable gauges to be made to check that the actual parts are within these limits. The other error being in the position of the apices of the two cones, he should give a linear dimension with a tolerance between one of the two previously mentioned reference diameters and a reference face, to limit the relative axial displacement of the two members. This dimension and tolerance cannot be measured directly and has to be gauged either with a "Go" or "Not-Go" cone gauge or such a device as a ball or ring and a depth micrometer.

Errors of truth, ovality, etc., can only be checked approximately by "clocking" but cannot readily be measured.

Fig. 59 gives an example of tolerancing on a taper mounted flywheel. The errors on the reference diameter have been assessed from experience. The tolerance on the "depth of engagement" of the two parts is determined to give correctly interchangeable assembly.

2. Position of Drilled Holes

As already indicated on page 52, holes should be toleranced to within a specified distance from their nominal positions. In practice,

with mating flanges the jigs for the two parts are often matched so that perfect interchangeability is achieved. If the holes are to be closer than 0·001 in. to their nominal position, jig manufacturing difficulties will be introduced. For press-fit bolts or dowels, this deviation should be allowed. For transition-fit bolts, allow 0·002 in. and 0·003 in. or more for clearance bolts. For holes which are drilled from marking off, witnout jigs, at least 0·02 in. clearance

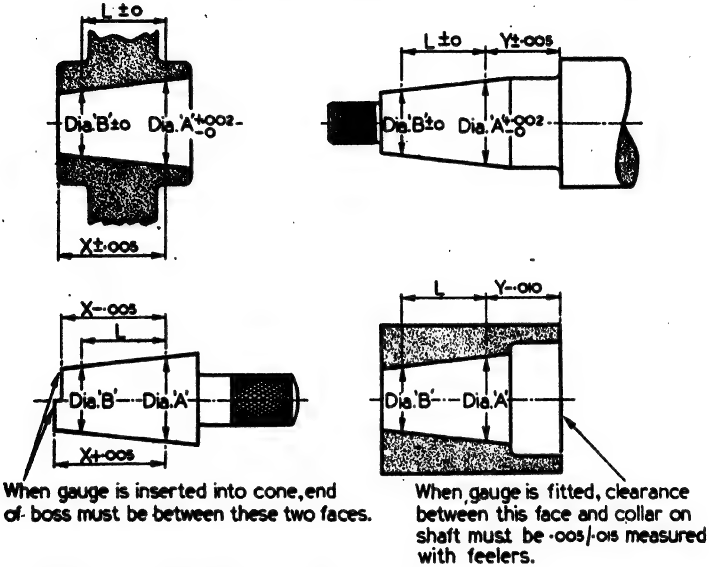


FIG. 59. AN EXAMPLE OF DIMENSIONING AND GAUGING OF A TAPER

should be allowed, preferably 0·03 in., on small holes, and up to 0·1 in. on a 1 in. bolt hole.

3. Splines

A spline is an interesting case requiring angular tolerances. Obviously if the two spline parts are to assemble correctly, the angle errors of the pitching of the slots and keys must be added to or subtracted from tolerances on the dimensions of the keys or slots. The problem has been avoided in British Standards by noting that "the spline should be able to be assembled in any position."

Fig. 60 shows the tolerance zones for a typical British Standard spline. The inner diameter of the hole, which can be produced accurately, locates the shaft radially, and thus this diameter and the diameter of the bottom of the shaft slots are treated as a normal

sliding fit, except that the clearance is to limit radial play more than to provide for lubrication. The top of the tongue slot in the hole is well clear of the shaft tongue tip and any convenient tolerances can be applied. The outside diameter of the shaft may be accurately controlled for other reasons.

The angular backlash is controlled by the relative widths of the slots and tongues and normal clearance fit limits can be applied,

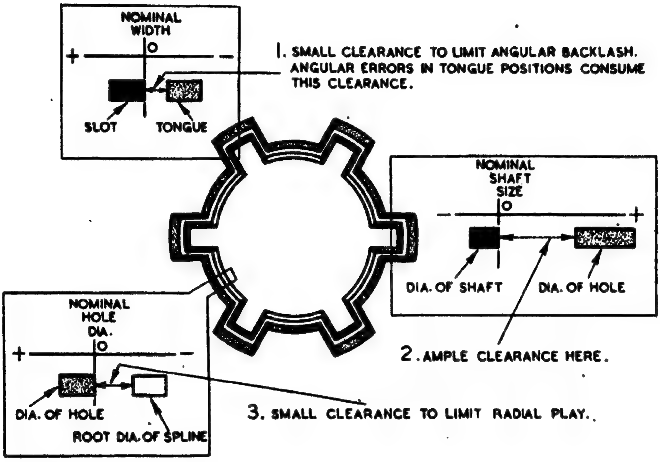


FIG. 60. SPLINE LIMITS AND THEIR TOLERANCE ZONES (B.S. 46)

the hole slot width being treated as a unilateral hole. Since angular pitching errors consume the minimum backlash clearance, and since eccentricity of shaft and hole affect the clearance as well, it will be seen that scientific tolerancing is almost impossible. Probably the best procedure is to call for a gauge with reduced radial clearance and slots perfectly pitched and on bottom tolerance as regards width, and then to specify the minimum backlash in any position, as measured with a dial gauge on the shaft tongues.

It is probably unnecessary in most cases to do more than assume that angle errors are not present and to specify, as in the British Standard (B.S. 46), that the shaft shall enter in any position.

Although normal hole and shaft tolerance systems should normally be applied to all spline dimensions, American standard involute splines have tolerances directly proportional to the diameter, and in the case of S.A.E. splines and serrations, the tolerances are substantially fixed and vary but slightly with the diameter.

The metric standard splines have been toleranced rather more scientifically as Table 7 will show.

TABLE 7

(I.S.A. Tolerance symbols are referred to)

	Location (bore of hole)						Location (o.d. of shaft)					
	Hole			Shaft			Hole			Shaft		
	W	*D*	*d*	*W*	*D*	*d*	*W*	*D*	*d*	*W*	*D*	*d*
Stationary assembly	H8 to H11	H11	H7	d10	d10	m6	H11	H7	H7 to H11	d10	w6	d10
Sliding assembly	H8 to H11	H11	H7	d10	d10	f7	H11	H7	H7 to H11	d10	f7	d10

W = width of tongue; D = outside diameter; d = inside diameter.

The tolerance on the width of the tongue should be determined taking into account the errors due to its length (the S.A.E. Standards quote a maximum out of truth of 0·006 in. per foot of the tongue with respect to the axis of the spline).

In the case of splines locating on the outside of the shaft, the bore of the hole may have to be made to a close limit (H7) anyway to guide the broach. In the other and more usual case, the hole has to be accurate for location and automatically guides the broach.

TOLERANCE ON PLATED PARTS

Some confusion exists in factories and drawing offices with regard to the correct tolerances to be applied to production components which are nickel or chrome plated to engineering limits, of which a typical example is the piston rod or tube used on all aircraft hydraulic jacks or undercarriage shock absorber legs. This component is either ground before plating and then lightly finish-plated by the so-called "flash" process, or is just heavily plated for final grinding. Each operation requires limits to be set to enable reasonable working tolerances to be allowed at each stage. The machine shop is usually given sufficient working tolerance, but often the plater is not allowed enough latitude and is expected to apply a nominal thickness of coating without a tolerance.

The various stages each require limits as follows—

(1) Initial turning or grinding operations require a tolerance of at least 0·0005 in., preferably 0·001 in. or more, according to the diameter. A tolerance of 0·001 in. for any diameter up to 4 in. should be obtainable without difficulty by grinding.

(2) The plating process requires (*a*) a minimum thickness of coating, (*b*) a maximum thickness which depends on whether

flash or heavy deposits are used. For flash chroming the coating thickness may be 0·002 in. to 0·003 in. on the diameter, i.e. a tolerance of 0·001 in. For heavy deposits limits of 0·006 in. to 0·008 in., or a tolerance of 0·002 in., will probably be satisfactory.

(3) The final part as assembled has its own working or assembly limits depending on its duty, and for a sliding fit may be, say, — 0·001 in. to — 0·003 in., i.e. a tolerance of 0·002 in.

From this it will be evident that three stages are to be dealt with, each having a tolerance of between 0·001 in. and 0·002 in. The first two operations must result in the final stage, and the limits fixed so that this is directly achieved in the case of flash chroming, or is possible by final grinding in the case of a heavy deposit.

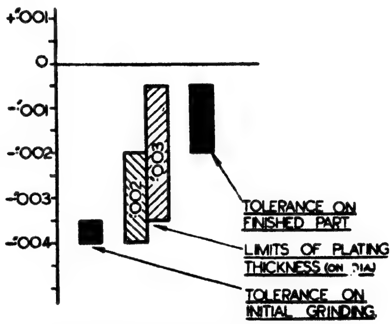


FIG. 61. "FLASH" CHROMING

An example of flash chroming will show this put into effect (Fig. 61). The finished part is required by the design to have sliding class limits of — 0·0005 in. to — 0·002 in., i.e. a tolerance of 0·0015 in. The plater requires a coating of 0·002 in. minimum and 0·003 in. maximum thickness on the diameter, this giving a tolerance of 0·001 in. The initial grinding process will, therefore, only have a permissible tolerance of the differences between these two tolerances, i.e. 0·0005 in., a point which is generally overlooked when discussing these questions. This can be understood from Fig. 61; the maximum initial size plus the maximum plating thickness gives the maximum final size, and the minimum initial size plus the minimum plating thickness gives the minimum final size.

It will be appreciated that starting with the initially ground part, whatever its exact size within the limits quoted, if a coating thickness of between 0·002 in. and 0·003 in. is added the final result falls between the limits desired.

This example presupposes that no selection or grading of sizes is carried out, and that all parts as ground are put in the plating bath together for sufficient time to give the necessary coating thickness. It is unreasonable to expect the plater to use selective plating, measuring the parts before immersion. Further, it is unlikely that the necessary personnel used to dealing with engineering limits in such a manner is available.

The example shows that if a tolerance of 0·001 in. on both

initial grinding and plating is required, the final toleranc one the finished flash-plated part must be at least 0·002 in. If initial grinding can be held to 0·0005 in. this can decrease to 0·0015 in.

If greater accuracy on the finished part is required, a final grinding operation will be necessary. In view of difficulties of truth and concentricity, it is not practicable to remove very thin layers of chrome and a relatively heavy layer must be deposited (0·006 in. to 0·008 in.), to permit the removal of a reasonable amount and also leave a sufficiently thick layer. The problem now is to find the correct limits to enable the various steps to be fitted together, i.e. an initial grinding operation with a tolerance 0·001 in., a coating of 0·006 in. to 0·008 in., giving a tolerance of 0·002 in., and a final grinding operation leaving about 0·003 in. of chrome for satisfactory wearing resistance, and to a tolerance of 0·001 in. or whatever is required by the designer.

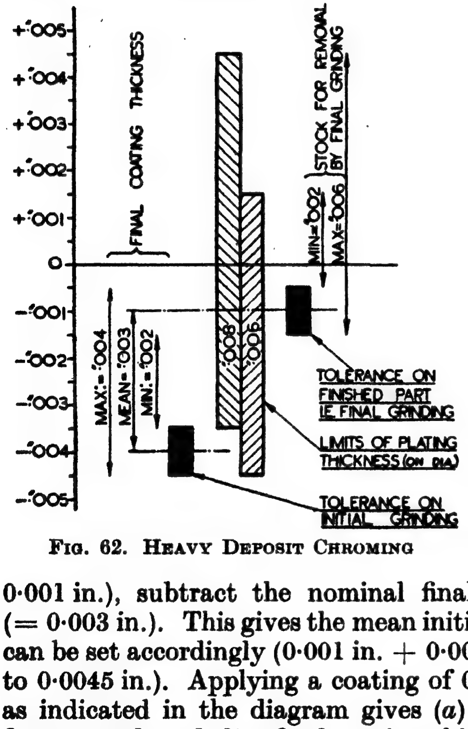


FIG. 62. HEAVY DEPOSIT CHROMING

Fig. 62 serves as an example, the process being as follows—

From the mean dimension of the finished part (0·0005 in. to 0·0015 in. = 0·001 in.), subtract the nominal final coating thickness required (= 0·003 in.). This gives the mean initial grinding size, and the limits can be set accordingly (0·001 in. + 0·003 in. = 0·004 in. = 0·0035 in. to 0·0045 in.). Applying a coating of 0·006 in. to 0·008 in. thickness as indicated in the diagram gives (*a*) 0·002 in. to 0·006 in. of stock for removal, and (*b*) a final coating thickness of 0·002 in. to 0·004 in., which in this case can be considered satisfactory.

Study of Fig. 62 shows that the maximum thickness of coating (0·008 in.) is not important (except that it affects cost), but that the minimum (0·006 in.) is critical. It is equal to the sum of (minimum layer to be removed by final grinding) + (final tolerance on part) + (minimum final or residual layer) + (initial grinding tolerance); i.e. 0·006 in. = 0·002 in. + 0·001 in. + 0·002 in. + 0·001 in.

This equation is of importance as it enables any particular case to be considered and the dimensions adjusted accordingly.

SELECTION TABLES

The following tables give summarized information on the selection of some of the simpler fits (on the hole basis), but the designer should not use these without having at least read pages 61–77 of this chapter. Cross-referencing between one system and another is only approximate and this should be borne in mind.

1. Holes

In certain of the systems more than one hole may be associated with a given shaft and therefore an introductory statement on hole limits is required so that the correct hole can be selected (see Table 8)—

TABLE 8

Hole				Remarks and Examples
System				
I.S.A.	B.S. 164	Newall	A.S.A.	
H5	—	—	—	Can only be used on lapped or diamond bored holes of the finest workmanship. Piston pin hole on light alloy engine piston.
H6	B		1	Obtainable by fine grinding or honing. Can be obtained by accurate hand reaming, but this is difficult and costly. Suitable only for very accurate bushes, bearings, spigots, etc., on precision production.
H7	—	A	2	Obtainable by grinding, honing, broaching, or high quality turning or careful reaming. Suitable for accurate bushes, bearings, spigots, cylinder bores on high quality production.
H8	U	B	3	Obtainable by all accurate production processes. General hole for good quality mechanical engineering.
H9	V	—	4	Suitable for coarse diameter fits only. Normal general engineering width tolerance for good quality milling, slotting, etc.
H10	—	—	—	Not for diameter fits; suitable for widths or lengths on unimportant mating parts.
H11	W	—	—	For very coarse fits only, such as bolts and washers, etc., general press work, etc.

2. True Press Fits

See Table 9.

TABLE 9

System				Remarks and Examples
I.S.A.	B.S. 164	Newall	A.S.A.	
p6/H7	UE	AD*†	Med. Force*	The standard true press fit of steel, cast iron or brass parts. Unlikely to damage or overstrain components, but requires a press or hammer for assembly. Also for the case when one part is in light alloy if a lesser grip can be tolerated (see below). Should not be used when components have to be readily dismantleable.
s7/H8	UE	AD	Med. Force	As above, but when one component is in light alloy, if the same type of grip is required.
n5/H6	BE	—	—	A substitute for the above when variation in interference must be kept at a minimum. Since extremely fine tolerances are involved this should not be used except under very special circumstances.
p5–x5/H6 r6–z6/H7 t7–z7/H8	UF	AF	Heavy Force	These fits cannot be used without thorough investigation, theoretical and practical.

* Becomes relatively heavier as diameter increases.
† BD in larger diameters.

3. Transition Fits

See Table 10.

TABLE 10

System				Remarks and Examples
I.S.A.	B.S. 164	Newall	A.S.A.	
n6/H7 n7/H8 m5/H6	DB DU —	— — —	Tight — —	These fits are on the average press fits of reasonable interference; only on extreme limits is there clearance. They can be used when *slight* clearance is no inconvenience, or when random selective assembly can be relied upon to eliminate the few clearance fits likely to occur. The ordinary bronze bearing bush fit is a typical application.
m6/H7 m7/H8 k5/H6 k6/H7 k7/H8 j5/H6 j6/H7 j7/H8	CB CU — BB BU — KB KU	— — — — — — — —	— — Wring-ing — — — — —	These are true transition fits where the amount of variation of clearance or interference is almost entirely controlled by the tolerances proper. Depending on the average interference or clearance required, so the fits m, k, j or C, B, K are used; m and C average slight interference; k and B average the same size approximately; j and k average slight clearance. The quality chosen (5, 6, or 7, etc.) depends on the quality of assembly required. m6/H7 or CB might be used on the casing spigots of an aircraft magneto; a bicycle crank and shaft might be j7/H8 or KU.
h/H	L	PA	Snug	This is the particular fit where the parts are only just size on extreme limits, and are otherwise clear. This is very widely used for non-running parts, the most common of all being bolts and nuts and screw threads (although the actual tolerances used are not from the standard limit systems). Other typical examples are pins in shackles, pins in holes retained by clamp screws, widths of a tongue and slot, general purpose spigots, etc. The selection of the quality of tolerance where variation is possible (I.S.A. shafts and holes and B.S. 164 holes only) is merely a question of the maximum clearance that can be tolerated.

4. Clearance Fits

See Table 11.

TABLE 11

System				Remarks and Examples
I.S.A.	B.S. 164	Newall	A.S.A.	
g5/H6 g6/H7 g7/H8*	— PB —	— — ZA	— — —	These are fits of genuine clearance, although it is small and is not sufficient for normal running. They are very convenient for precision push fits of delicacy and for this reason g7, etc., are not recommended in the I.S.A. system for general use, as being too wide in tolerance. An example would be a precision hinge or articulation pin subject to occasional rotation, where play is to be avoided. Another is on readily detachable spigots.
— f	M Q	— Y	— Med.†	These are the first true running fits. All can be safely used on all normally lubricated pins, shafts, etc., either grease or oil lubricated where speeds are low and temperature difficulties are not likely to be encountered. They cannot be recommended without qualification for heavy duty bearings.
— e	— R	X —	— Med.‡ Free§	These are similar running fits but rather slacker. They can be used on less important pins and shafts but only where the additional maximum play is not an inconvenience. In the case of heavy duty bearings the controlled slackness of the fits e7/H6, RB or XA may be useful.
d, c, b, a	S, T, TT	—	Free, Loose	It is not possible to recommend these fits in a general selection table, because their use will depend on numerous factors to be predetermined accurately. They must be used as the case merits.

* Use not recommended.
† Small sizes only (f7).
‡ Larger sizes only (e7).
§ Small sizes only (e8).

EXAMPLE OF LIMIT AND TOLERANCE APPLICATION

In Fig. 63 is illustrated a high-pressure hydraulic pump for aircraft purposes; a detailed study of the various problems confronting the designer in setting tolerances on this pump should be of practical value. Before attempting to explain the limits adopted, the operation of the pump should be appreciated.

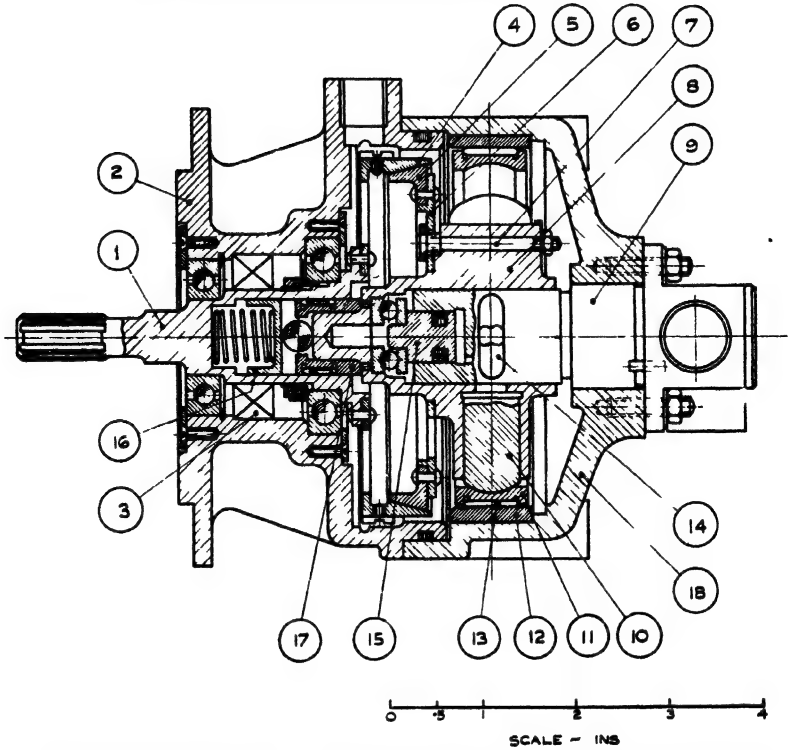


FIG. 63. A HIGH PRESSURE HYDRAULIC PUMP USED TO ILLUSTRATE THE TOLERANCE SETTING PROCEDURE

The splined drive shaft (1) is carried in the front aluminium alloy casing (2) on two ball bearings and sealed against ingress of oil, etc., by an oil seal (3). On a circular flange on the inner end of the drive shaft is riveted the female member (4) of a cone clutch. (This member carries several small studs or buttons which centrifuge any oil inside the pump back to the tank.) The male clutch cone element (5) is a cast-iron ring riveted to a carrier plate (6), located radially on the bronze pump rotor, and driven by several headed pins (7). The pump rotor (8) rotates on a hardened steel rotor shaft (9) which

is provided with drilled and tapped inlet and outlet ports. The rotor also has five radial cylinders in which operate five solid steel pistons (10). The tips of the piston contact the inner curved face of the inner ring (11) of a needle race, the outer fixed ring being (12) and the needles (13). When the pump rotor turns, the pistons fly out under centrifugal force and the needle race, by rotating, prevents excessive wear at their ends. The needle race assembly is slightly eccentric (0·1 in.) with the pump rotor axis so that the pistons as they rotate reciprocate with a stroke of 0·2 in. The oil is sucked and delivered through the longitudinal passages in the rotor shaft and in and out of the cylinders through two slots milled across the shaft (one is seen at (14)), the position of the slots being arranged to give the necessary timing. The pump, which develops 3000 lb./sq.in., relies on the accuracy of manufacture and closeness of fit of the cylinder rotor bores and the pistons and shaft diameters, for the necessary sealing.

A further passage down the rotor shaft admits oil pressure (controlled by a special valve device in the hydraulic system) into a cylinder at the end of the rotor shaft and by moving a glanded piston (15) applies the necessary force to the male clutch member, by moving the rotor assembly bodily towards the female clutch cone, to enable the clutch to transmit the drive torque. When the pressure is released the clutch releases and the pump proper ceases to rotate; the larger central spring, spring guide (16) and ball assist release. The outer end of the rotor is carried on a needle race assembly (17).

The rotor shaft is carried in and bolted to a further casing (18), which is clamped to the front casing and sealed by a rubber ring gland.

1. Fits of Fundamental Importance

Without special fits on the rotor assembly the pump could not work; this then is the first problem to tackle. The rotor holes must be diamond bored with very carefully designed spindles and a first-class machine; the limits set are + 0·0002 in, (0·005 mm), + 0, on all six main holes. (I.S.A. H4; special on other systems). The shaft main diameter and pistons must be hardened, ground, and lapped to achieve the necessary accuracy ("superfinished" in the modern terminology). Particular care must be taken if centreless grinding is adopted that triangulation or lobing is not in evidence. The limits set are $\pm$ 0·0001 in. ($\pm$ 0·003 mm). The final fit is obtained by random selection to give about 0·0001 in. (0·0025 mm) clearance. If the factory processes and equipment are up to it, even better results and quicker assembly would be obtained by working to + 0·00015 in. (0·003 mm)/+ 0 = (H3), for the holes and

— 0·0005/— 0·00015 in. (0·001/0·004 mm) for the shaft, although the very highest precision is required in this case. With the less precise fit, the parts are assembled slightly tight and run in.

Accuracy of circularity and parallelism well within these limits must be maintained.

2. Other Limits

The actual pumping fits settled, the other limits on the main parts can be considered.

Pistons (10)

As 0·05 in. nominal clearance is allowed at the bottom of the piston bores, a general length tolerance will do. The spherical top radius should be limited so that central dome contact is always obtained, i.e. — 0, — 0·010 in. on the piston and + 0·010 in., + 0, on the inner ring (11).

Rotor (8)

The angular spacing between the five cylinder bores is not important and can be dimensioned as $72° \pm 1°$, this being readily obtainable by the indexing fixture which will be used for the bores. No attempt is made to tie up the truth or coincidence of the five cylinder bore axes, as this is relatively unimportant. If it had been important, and in view of the fine tolerance on the bores, a check by means of mandrels inserted in the bores could be specified, the mandrels when inserted having to fit over sets of pins, to check angle errors, and a dial or height gauge test being used to check the accuracy in the plane across the rotor shaft hole.

The depth of the bores in relation to the axis being unimportant, an open dimension is sufficient. The registering diameter or spigot on which the male clutch element (6) sits is not to be too close a fit, to allow the clutch to centre slightly, and a fit with a minimum clearance of 0·001–0·0015 in. is used. Excessive clearance may let the clutch rub when free (H8–f7 fit will do). The five clutch driving bolts (7) can be fitted in normal reamed holes, say (H9–f8) giving hole + 0·0013, + 0 in. and shaft — 0·00045/— 0·00125 in.; the pin diameter can be centreless ground to a fit within the thread tolerance (e.g. about + 0·0005/+ 0 in. = h7).

The holes in the rotor should be drilled from the locating end (in case the holes run out) and can be held to + 0·002/+ 0 in (say H10). Alternatively, the holes can be drilled clear from the other end for most of the depth and a short length of fitting diameter at the important end can be reamed to a good fit (H8 say). The accuracy of spacing of these holes at the important end should be

to within 0·001 in. of the nominal position with fitted pins, or 0·002 in. with small clearance pins.

To limit the endwise play of the clutch plate (6), the two faces on the rotor which affect it should be held to — 0, — 0·002 in. by a suitable dimension. The head or shoulder of the pin (7), should have a length dimension controlled to + 0·002, + 0 in.; finally the clutch plate (6) should have its thickness (the inner part only) held to — 0·002, — 0·006 in. This gives an end float of 0·002/0·010 in.

To control the axial position of the rotor ports and bores, they should be dimensioned off the clutch face referred to above, but the concentricity of the ports and bores need not be held to better than 0·010 in. dial reading. With this generous tolerance, a concentricity plug gauge may be used, as the tolerance on the bore can be ignored.

The depth of the rotor shaft bore must be well clear of the end of the shaft, so that with the clutch withdrawn and disengaged, the thrust race fouls the end of the shaft before the rotor does. Particular care must be taken that a tight place is not left at the bottom on the rotor bore and a generous but shallow undercut is an advantage, the fine boring tool running into it (this is not shown on the drawing). The width (or height) tolerance of the thrust bearing can be obtained from the bearing maker's catalogue and is about ± 0·002 in. The flat face in the rotor on which the bearing seats must be true and square with the rotor axis, and its depth can be dimensioned from the outer end of the hole to + 0·010/+ 0 in.; the position of the undercut in the main bore already referred to should be dimensioned with a similar tolerance from the outer end of the hole, making the total variation between the end of the undercut and the outer face of the thrust race ± 0·002 — (+ 0·010/+ 0) + (+ 0·010/+ 0) = ± 0·012 in. A nominal difference of about 0·08 in. will be sufficient.

The recess diameter in which the bearing sits would normally be machined with a close locating fit (— 0/— 0·002 in. on bearing outside diameter; + 0·002/+ 0 in. on hole), but in this case the bearing is always maintained in contact by the spring assembly (16) and a clearance fit will do (say + 0·010, + 0·005 in.).

The small diameter of the rotor on which fits the needle race must be true and concentric with the main hole, and in fact may be ground to size off it. The needle race must be a true press fit, and a suitable fit is (H6 — n5), giving + 0·0004/+ 0 in. as the hole and + 0·00075/+ 0·00045 in. on the shaft.

As regards length dimensions, no special attention is required other than proper choice of reference faces, as there is ample clearance. As has already been mentioned, the face which locates the

rotor axially is the seat of the thrust race; care must be taken that there is ample clearance on the end of the rotor hole face so that it will not foul the inner face of the casing. The tolerance on this clearance is $\pm$ 0·012 in. as before, plus variations in the rotor shaft length and the casing boss width. With the limits for these suggested below the total variation will be $\pm 0{\cdot}012 \pm 0{\cdot}010 \pm 0{\cdot}010 = \pm 0{\cdot}032$; obviously a substantial nominal clearance must be allowed.

Rotor Shaft (9)

As mentioned above, the length of this part is important. The flange should be specified as true and square to 0·001 in. dial reading with reference to the other two main diameters, themselves concentric to 0·001 in. dial reading, and the overall length to the end from the flange should be $\pm$ 0·01 in. The length of the spigot diameter fitting in the rear casing should be $-$ 0·010/$-$ 0·020 in. below the casing minimum width so that this does not affect the rotor clearance; better still make it 0·05 in. short.

The diameter of the shaft spigot and the fit in the housing must be very close. A light tap fit is acceptable and a lower transition fit is suitable. (H6 $-$ k5 $=$ + 0·0006/+ 0 on the hole and + 0·00055/+ 0·0001 in. on the shaft). As the retaining studs can hold the shaft by friction, the stud holes can be normal clearance holes.

The inner bore forming the clutch operating cylinder carries the piston (15), which, as it has rubber sealing rings must have a fit suitable for these. This depends on the design of gland, but H9 $-$ f8 would do in some cases (hole + 0·0018/+ 0 in.; piston $-$ 0·0007/$-$ 0·0018 in.). A smooth honed or polished finish in the bore is usually required. The depth of this hole must be such that the rubber gland does not jump into the undercut when the piston is fully home (assuming the thrust race is pressed on firmly); two dimensions can tie this up and therefore a nominal 0·05 in. ($\pm$ 0·010 $\pm$ 0·010 $=$ 0·03 in. land minimum) will do.

The concentricity of the piston bore with the main shaft diameter is important, otherwise the thrust race may be located eccentrically, but in view of the possibility of slight self-alignment of the piston, a tolerance of 0·002 in. dial reading should do.

The accuracy of milling the cross slots (14) is important, as port timing may be affected, and these should be dimensioned off some convenient reference face, not, however, the shaft flange, as this is ground after the slots are milled. The inner end of the shaft will do as it is not ground after hardening and the flange can be ground up off it. Two dimensions each $\pm$ 0·010 in. from one side (not the centre) of the slot to the shaft end and back to the flange will locate

the slots to $\pm$ 0·020 in. The nominal slot position obviously should be correct with the clutch engaged.

Rear Casing (18)

This carries the rotor shaft in a bore already dealt with (i.e. H6 = + 0·0006/+ 0 in.). The end face, however, is just as important to maintain the truth of the shaft, and it should be true and square with the hole and with the other parallel face. The large spigot diameter between the two casings should be an "assembly with least play" fit and (H7–h6) is suggested—hole + 0·0014/+0, shaft − 0/− 0·0009 in. Since for ease of disassembly this is kept slightly slack, the main accuracy is derived from the end faces. The two faces of the rear casing, therefore, should be specified as parallel within 0·0005 in. dial reading, with the gauge used on the smaller or shaft flange end. Incidentally the stud holes should be recessed to prevent the first thread pulling up and spoiling the face. It is not easy to specify the accuracy of the shaft spigot hole in relation to the flange face, as the tolerance on the hole limits the accuracy of the gauging, if a mandrel is used. A gauge equivalent to a dummy shaft would be useful but for more accurate checking the housing would have to be held with the flange face on a true vertical plane, and a dial gauge used along the hole axis, either directly on its rather short length or using a well-fitting mandrel. The drawing should specify the error on the extremities of the hole with respect to the flange face, a suitable error being 0·0005 in. It is important to note that in all cases of this type the flange exerts more influence on accuracy than the diameter, unless the latter is a true press fit.

The fit of the outer needle race ring (12) in the housing, the ring not being keyed, must be sufficiently tight to remain so at elevated temperature; the estimated maximum design temperature being 100° C., a slight minimum interference is required at this temperature. Since the casing is aluminium alloy and the ring of steel the differential expansion of casing* at 100° C. (i.e. 80° temp. rise) will be $0{\cdot}8 \times 4{\cdot}0 \times (0{\cdot}0022 - 0{\cdot}0013) = 0{\cdot}0029$, say 0·003 in. If we accept a tolerance on the hole of H7 (= + 0·0014/+ 0 in.) the lower limit of the shaft will then be $0{\cdot}003 + 0{\cdot}0014 = 0{\cdot}0044$ in. The nearest standard fit is t (= 0·0042 in.) or u (0·006 in.). The fit t6 (+ 0·0051/+ 0·0042) can be assumed as satisfactory for a practical trial.

With this tolerance the maximum interference will still allow the parts to be assembled easily if the casing is heated in boiling water or hot oil to 100° C.; the ring can then be dropped in. To

* See Appendix 3.

extract, the casing will have to be heated to a much higher temperature, since the ring heats up as well.

The rotor slides on the shaft during clutch engagement and takes the inner needle race and needles with it. The nominal position of the outer ring should therefore be central with the clutch engaged with a small overlap of width on either side for disengagement, or wear on the clutch cones. The dimension on the casing from the bottom of the race ring diameter to the outer end of the casing controls this.

The bore carrying the needle race is eccentric with the rotor shaft hole by 0·10 in. A jig will be needed to establish this and there will be two main sources of error, the dimension of eccentricity and the angular position of eccentricity in relation to the remainder of the casing. A tolerance of $\pm$ 0·005 on the eccentricity will affect pump output by a theoretical $\pm$ 5 per cent, which may be acceptable. No difficulty should be experienced in holding the eccentricity to $\pm$ 0·002 or, say, a travel of 0·2 $\pm$ 0·004 as measured with a dial gauge on the shaft locating bore, the inner needle track being rotated on a mandrel; this limits the output error to $\pm$ 2 per cent. The angular position of the eccentricity (i.e. "top dead centre") should be controlled in relation to the flange stud holes, and in this particular case is not critical. The top dead centre as indicated by a dial gauge should be correct $\pm 1\frac{1}{2}^\circ$ or about $\pm$ 0·05 in. measured on the edge of the needle race, and, for convenience, with respect to the main casing holes. Incidentally, the most accurate method of getting the top dead centre is to mark the two positions giving a certain dial reading and then take the mid-point.

The castings are bolted together with four through bolts in clearance holes, the holes being pitched accurately to within 0·002 in. of their nominal positions in relation to each other, but 0·005 in. with relation to the four stud holes; the latter can be correct as well to 0·002 in. amongst themselves; finally the small peg, which allows the shaft to be fitted in one correct position only (alternatively, the studs could be asymmetrical although this makes impossible reversing of the pump by turning the shaft through 180°) can be considerably clear in its hole in the casing, say 0·01 in. nominal with the two holes for it correct to 0·004 in. on shaft and casing. The total angular error of the rotor shaft in the rear casing would be equivalent, apart from clearances on the holes and bolts or studs, to 0·005 in. on each of two pitch circles, or a total of about 0·018 in. on a p.c.d. of 1·5 in. = 8 minutes of a degree.

Outer Needle Race (12)

The outside diameter and width of this has already been settled.

It may well be that the inside diameter distorts so much on assembly in the casing that the inner diameter cannot be finish ground before insertion. Assuming this is so, the ring would be ground eccentrically in position after assembly, using the shaft spigot hole as a location, and many of the remarks just made about the eccentricity apply at this stage too.

The limits on the inner diameter of the outer ring and the outer track of the inner ring determine the fit of the needle race assembly. The tolerance on the needles (13) is given by the makers, if standard needles are to be used, but can be taken as $\pm$ 0·0001 in. The minimum reasonable tolerance on the outer bore is H5 (+ 0·0006/ + 0 in.), although H6 is more practicable (+ 0·0009/+ 0 in.). The equivalent tolerance on the outer track of the inner ring is again Quality 5 or 6 (0·0006 in. or 0·0009 in.). Assuming Quality 5 on the shaft and 6 on the hole, the total variation in fit is $0{\cdot}0009 + 0{\cdot}0006 + 2 \times 0{\cdot}0002 = 0{\cdot}0019$ in. Unless a complicated selective assembly process is to be used, the needle race assembly will not be a very precise fit, but fortunately this type of race works better with appreciable clearance than without it. A minimum clearance of 0·002 in. is found by experience to be satisfactory, with a maximum in this case of 0·0039 in.; this maximum can be exceeded somewhat at the expense of a slight amount of noise and it may be that the following tolerances will be satisfactory—

Needles	$\pm$ 0·0002	Tolerance: 0·0004 $\times$ 2
Hole H7	$^{+\,0{\cdot}0014}_{+\,0}$	Tolerance: 0·0014
Shaft h6	$^{-\,0}_{-\,0{\cdot}0009}$	Tolerance: 0·0009
Total variation:		0·0031 in.

Working clearance: 0·002/0·005 in.

The exact diameters involved depend on the number of needles and the clearance between them, and the nominal values should be calculated, according to the needle maker's rules.

Inner Needle Race (11)

All relevant dimensions have been discussed already, except the width of the needle track and the question of ovality. The needle track can be about 0·010 in. wider than the needles which are held to reasonable limits. Two small undercuts at the edges on the needle track eliminate fouling of the ends due to the radius on the grinding wheel.

As the ring is comparatively thin, trouble will certainly be

experienced with ovality. This need not be held to the normal limit within the tolerance (about 0·0009 in.) and as long as the outer track is not more than 0·0015 in. oval, and provided the race has the 0·002 in. minimum clearance mentioned above, the assembly will function satisfactorily.

Front Casing (2)

The spigot of the front casing in the rear casing has already been discussed. The truth of the face at the end of this spigot is again of paramount importance. The two diameters in which the ball races sit are important in relation to each other and to the spigot. Their diameter tolerances can be fixed from ball bearing makers' information, depending on the type of bearing used, and one bearing housing should be true with the other and with the spigot when the casing is fitted to an accurate dummy female spigot; the dial gauge errors on these two diameters should not exceed 0·002 in. if the errors are in phase or 0·001 in. if out of phase.

The casing is designed so that the front mounting flange is machined first, and then using this as a location, the central holes and the spigot are machined at one set-up.

The diameter which carries the oil seal has to be machined to a reasonable tolerance (e.g. H8 = + 0·0016/+ 0 in.) and can be slightly eccentric with the bearings as the oil seal is elastic. If the eccentricity exceeds 0·004 in. (dial reading) leakage may occur.

If the inner face of the front flange carrying the front bearing retaining plate is used as a length location for machining the holes and spigot, as is intended, the length dimensions locating the bearings will be readily controlled. The circlip groove should be dimensioned between the reference face and the farthest edge of the groove. The bearing end float is then this tolerance, plus the tolerances on the circlip width and bearing width. In this case probably + 0·005/+ 0; − 0/− 0·002; − 0, − 0·002 in. respectively; this gives 0 to 0·009 end play, which is acceptable. The circlip position can be readily held to this tolerance by using a special setting bar or stop; normal capstan stops will not do.

The position of the oil seal is not so critical that normal dimensioning cannot control it; the larger bearing housing, however, can be dimensioned as before with two dimensions from the reference face, the difference of which gives the width of the recess to be compared with the bearing width. As this is the thrust location bearing, the outer ring must not have any end play, and therefore to get reasonable control over the bearing without bending the plate which retains it, or having to use shims, it will be better to dimension the width of the recess directly, and rely on the production engineer

using a combined tool for both its faces; an open dimension can position one face and the tolerance between the two faces can be 0·002 in. Since the bearing width tolerance is 0·002 in., this gives a maximum bend of the plate of 0·004 in. if the housing and bearing are "size and size" on lower limits.

The final important length dimension is the one which locates the rear casing with respect to the female clutch through the large ball race. If the dimension from the face of the spigot to the reference face at the other end is used, then the tolerance on it has to be added to the tolerance on the dimension locating the large ball race recess, suggested above to be "open" (i.e. $\pm$ 0·010 in.). This may be excessive and a single dimension (say $\pm$ 0·005 in.) from the spigot face to one of the ball race housing faces will limit it to 0·010 in. plus a slight variation in the width and position of the race. This is not an easy dimension to work with and it may be better to limit the earlier one to $\pm$ 0·005 in., with a similar tolerance on the dimension from spigot face to reference face. The variation then is about 0·02 in.

The flange mounting dimensions are to some appropriate standard, the spigot diameter being true with the bearing bores to 0·001 in. dial reading.

Drive Shaft (1)

The driving splines are also to some appropriate standard (e.g. B.S. 46). The two bearing diameters should be ground to the correct ball bearing shaft limits, the first fit being a sliding one to allow for differential temperature expansion due to the light alloy casing. The diameter on which the oil seal runs should be determined in the first place by the core diameter of the bearing retaining nut, but oil seals are only available for certain common fractional or millimetre sizes. The tolerance on this shaft can be h9 (— 0/— 0·0029 in.) and best modern practice is to call for plunge grinding so that any small scratches are circumferential not helical, thus reducing the abrasion of the oil seal. A "super-finish" would be even better. The top diameter of the thread whatever it may be according to the thread tolerances, should be ground to the same limits as the bearing diameter close to it. If the bearing is metric, the thread may conveniently be metric too.

The remaining diameter of importance is the locating diameter registering the female clutch cone (4). The fit here should be a lower transition fit (e.g. H7 — j6; hole = + 0·001/+ 0 in. and shaft = + 0·0004/— 0·0002 in.). With this fit, in the worst case the cone will need tapping on, but on the other hand the clearance is limited to 0·0012 in.

The accuracy of the spline outside diameter, the two bearing diameters and the clutch register can easily be checked with the shaft spun in the inevitable centres. In all cases the run out should be 0·001 in. dial reading maximum, although if the clutch cone is to be ground after riveting to the shaft (see later) its register may be out 0·004 in.

The flange side on which the clutch cone is riveted should be true and square under the same test to 0·0005 in. or 0·001 in. if the clutch is to be ground up later.

Since the shoulder on to which the inner bearing is clamped is the most important end face, it should be ground true and square, but 0·001 in. dial reading should do, as the ball bearing has slight play. This face should be used also as a length dimension reference, controlling the position of all other important external lengths. A single dimension should position the clutch cone disc, with a tolerance (since both faces are ground) of -0, $-0{\cdot}005$ in. The relation between this flange on the shaft and the front casing spigot is therefore controlled to $0{\cdot}020 + 0{\cdot}005 = 0{\cdot}025$ in.

The internal dimensions of the shaft are unimportant as regards length, but the main bore should be machined and ground accurately to form the outer track of the needle race (17) as well as a pilot for the spring cup (16). Exactly the same remarks regarding the needle race tolerances apply in this case as on the larger one, except that there must be no appreciable running clearance, otherwise the whole purpose of the needle race is nullified. This question is dealt with below.

Several dimensions control the clearance between the spring cup (16) and the bottom of the hole in the shaft, and a nominal 0·15 in. is needed to allow for this and clutch wear.

Needle Race (17)

The fit of the rotor projection and the bore of the needle race has already been settled as (H6–n5) giving an interference of 0·00075/0·00005 in. Since a precise fit is required in the needles, they must be to the most accurate limit—i.e. $\pm\,0{\cdot}0001$ in. The bore of the shaft can be ground or honed to H6 limits ($+\,0{\cdot}0005/+\,0$ in.), and the outer track of the needle race to Quality 5 tolerance (0·0003 in.). This gives a total variation of $0{\cdot}0005 + 0{\cdot}0003 + 2 \times 0{\cdot}0002 = 0{\cdot}0012$ in. Some small amount of the 0·00075 in. maximum interference will be passed on in expanding the needle track and this may upset the clearances slightly on extreme limits. The best thing, however, is to ignore this and use the limits indicated; the expansion of the needle race will reduce the mean clearance in the average case, and selective assembly will reject a race or

some other part on extreme limits. By this means the absolute maximum clearance is about 0·00125 in., which is adequate to support the rotor and prevent distortion of the rear casing under load.

The outer diameter of the race can be machined clear of the shaft bore by about 0·005/0·010 in. The width of the track should be similarly wider than the needles. The hole for the ball is not important as the ball will centralize automatically; a normal clearance size will do.

Spring Cup (16)

The outside diameter can be held to slack running limits (e.g. f8 or e8). The wall thickness of the disc if given by two open limits may vary $\pm$ 0·02 in., and this may be excessive. Two unilateral limits can limit it to + 0·020, + 0 in.

Female Clutch Cone (4)

This is riveted to the shaft, and as already mentioned it may be preferred to grind up the cone face after riveting. This is not necessary if adequate limits are set to the two individual components.

The central spigot between the cone disc and the drive shaft has already been settled as (H7 — j6). The disc wall thickness requires controlling to reasonable limits, about $\pm$ 0·005 in., so that the stiffness of the clutch once determined as satisfactory is duplicated in quantity production.

The only other dimensions which are at all important are those controlling the cone; this will be ground up, using the central hole as a radial location and the face equivalent to the flange of the drive shaft as a location for true running. Actually the best way to ensure accuracy of this face is to make it stand proud of the main disc of the clutch member by 0·010/0·020 in., for a diameter equal to the flange, and to grind or fine turn it to size when doing the bore. The cone angle is deliberately made less by $\frac{1}{2}°$ or 1° than the male cone member to eliminate clutch grab and therefore a tolerance of say $\pm$ 10 minutes can be allowed on the angle. This can easily be checked as being within two cone gauges. A particular diameter of the cone must be controlled to limit the nominal position of the apex of the cone, and thus the engagement of the male member. The most convenient method of gauging this is to use a cone gauge (the gauge angle being as large as the tolerance will allow) and to set limits on its engagement, the limits being represented by the end face of the gauge and a recessed step on it, as the "Go" and "Not-Go" faces, between which the edge of the cone must lie. Thus the drawing should first of all limit the overall width of the

cone to, say, + 0·005, + 0 in., and then demand that a certain specified theoretical diameter shall not enter the cone by more than, say, 0·010 in. nor fall outside it. Another way would be to give the outer diameter of the cone with a plus tolerance, but this means that someone has to calculate the equivalent penetration of the gauge. The tolerance on the position of the theoretical cone apex with respect to the thrust location bearing is therefore 0·005 + 0·005 + 0·010 = 0·020 in.

The rivet holes in the clutch cone and drive shaft flange can be drilled clearance holes, each within 0·002 in. of their nominal positions, and the squeezed rivets will fill the holes and cover any slight lack of register.

The accuracy of the cone track when the cone is assembled on the shaft and the latter rotated on centres should be within 0·002 in. dial reading; this test also covers wobble of the clutch face. This accuracy should be obtainable without final grinding, but if not, or if it is considered cheaper to do this, there can be no objection on technical grounds.

Male Clutch Cone Assembly (5) *and* (6)

The clutch carrier plate (6) has already been discussed. The male element (5) of cast iron should be riveted and the cone ground up with the disc located as for the female member, on the bore and local face around the bore.

A single dimension should give the length between the rotor face of the clutch disc and the outer or inner ends of the cast-iron clutch element, again to limit the movement of clutch and rotor. The most convenient dimension happens to be the overall width of the cone assembly and this can be held to — 0, — 0·005 in.

The angle of the cone should be toleranced as before, and the check on engagement (i.e. the apex position) is again given by a cone gauge (in this case a female gauge whose angle is at maximum) with an opening cut away and stepped to show whether the small end of the cast-iron cone overlaps the step limits. The drawing should specify that the theoretical diameter of the end of the cast-iron cone should lie on the end or not more than 0·010 in. along the part.

These cone tolerances control the reference face of the rotor to within 0·010 + 0·005 = 0·015 in. of the theoretical position of the apex.

The truth of the cone with respect to the central hole should again be within 0·002 in. It may be easier to check this with the cone held and a dial gauge in the bore.

The rivet holes between the two parts should be dealt with as before. The five holes driving the clutch may be clear of the pins

but they should be reasonably to size (say H9 = + 0·0013 in./+ 0) and pitched to 0·002 in. so that the drive is shared by the pins and not thrown out of true by them.

Clutch Driving Pins (7)

The limits on these pins have already been discussed under the rotor. The two diameters of the pin should be concentric to 0·001 in. dial. The width of the pin head has already been settled. The overall length (or better the length to the first shoulder) should allow a reasonable length of thread for the nut and locking plate, but because of the latter, the tolerance may be more than with a split pin hole.

Clutch Operating Piston (15)

The fit of the piston in the rotor shaft has been discussed. The groove dimension limits depend on the type of seal used but will not be close. The spigot on to which the thrust race is pressed should be to bearing manufacturer's requirements, although an upper limit in excess of the normal will do little harm in this case.

3. Conclusion

The foregoing analysis does not pretend to be exhaustive, but may be of interest as the pump introduces various problems of tolerancing and manufacture which are seldom combined in a single unit of this relatively small size and simplicity. The various limits chosen have deliberately been described instead of illustrated, so that the reader may work the solution out for himself if it is not entirely clear from the description.

Even this survey will have involved the careful reader in some study; it will be appreciated that in the actual design of a component, to dimension and tolerance all drawings will take some considerable time and thorough investigation. It is not surprising that many designs (perhaps most designs) do not get this detailed study, and the result is that difficulties are experienced in production, particularly with length or clearance tolerances, the more obvious diameter tolerances being corrected usually in the experimental or preproduction stage. Since trouble is usually due to combinations of adverse circumstances, it may be quite late in production before some troubles occur. The author recalls one case where after about 20,000 aircraft shock-absorber assemblies had been made over a period of five years, trouble in service was suddenly experienced. On investigation it was found that a diaphragm fitted into the shock absorber cylinder had the wrong limits on the drawing and would allow excessive distortion of the cylinder when clamped up. The

reason why it was suddenly found out was because a new process had been adopted which tended to keep all cylinders on the opposite extreme limit to previous practice; the small percentage of trouble experienced in the past when the faulty limit was reached had been considered average and had not been investigated in detail.

No effort should be spared to correct and improve drawings, particularly by adding simple notes in regard to accuracy (or "macro") errors. Designs and drawings should be under constant review during production so that they can be as up to date as possible.

CHAPTER III

QUALITY CONTROL—ITS INTEREST IN TOLERANCE SETTING

1. Introduction

"Quality Control" is a popular name for a process involving the application of statistical theory to the interpretation of inspection results. Although not a new process, the interest in maximum efficient production during wartime has done much to spread knowledge of its principles, this having been facilitated by several government sponsored "Quality Control Advisory Panels," and various publications (such as B.S. 600R).

In many engineering shops true statistical methods cannot be applied, and in fact it can generally be said that only relatively few mechanical engineering shops can make full use of the process; its outstanding importance, however, is that it has led to a much better understanding of the significance of limits. As will be seen later, if the tolerance of a properly controlled repetition process *is halved, only* 11 *per cent of the parts will be outside the new limits.* This remarkable fact is becoming more widely appreciated, and already there are signs of a few intelligent designers actually widening limits on certain parts, having considered the *probability* of the parts being inside the original limits, and the relative infrequency of the extremes of the new limits actually occurring.

Before, however, such revolutionary ideas can be made use of, the theory and practice of Quality Control, and particularly its limitations, must be thoroughly understood.

2. Conditions under which Theory of Probability can be Applied

If a machine is set up to do a given operation (e.g. centreless grinding of a batch of piston pins), there are inevitably, apart from wear of the cutting tool, grinding wheel, etc., various factors which affect the size of the component coming off the machine (distortion, temperature effects, surface deformation of the raw material, etc.). Therefore, the parts will differ from the set size, being either slightly larger or slightly smaller. This effect is considered in detail later.

The influence of tool wear, however, is of considerable importance and under certain circumstances can entirely preclude the application of probability theory; an example of this is in the fine boring of small holes, when the tool may suddenly break down and require sharpening.

If the tolerance is small and approaches the errors due to machine variables, very careful attention to the machine may be necessary, and although quantities may be large, the operator's attention may upset the distribution of sizes of the components. An example of this is precision cylindrical grinding using a "sizing" attachment; as the operator controls the final size according to a visual indicator, he has an influence on the probability of the part being any particular size.

The extreme example is in "free-hand" or batch production, when the operator almost entirely determines the final size, depending on his skill and the measuring equipment at his disposal, and often all the parts may be bunched together at one extreme of the limits instead of being spread according to chance and probability.

There are accordingly certain conditions which have to be met before Quality Control can be operated with success—

(*a*) The process must be one where the influence of such factors as the rate of tool wear or the operator's skill are of relatively minor importance and the machine itself exerts the main influence.

(*b*) The quantities must be large enough for the probability of an average being achieved being high—at least 100 pieces, preferably at least 1000.

(*c*) The means of measurement must be such that the true size of a component can be obtained accurately to within at least one-tenth of the work tolerance—it is no use trying to apply Quality Control to precision grinding with tolerances of 0·0005 in., using micrometers.

(*d*) As a corollary of this the tolerances must in general be fairly generous (preferably not less than 0·002 in.).

(*e*) Finally, a sufficiently intelligent staff must be available to understand and operate the system. Judging by the complication and abstruseness of textbooks and official publications on Quality Control, few inspectors can be expected to operate the system without special training.

3. The Theory of Probability

As already mentioned, if a machine is set up for quantity production of a given sized component it will, if properly under "control," produce parts which are either slightly larger or slightly smaller than the set size. On theoretical grounds it can be shown that if the parts are measured and graded into, say, eleven equal steps within the tolerance the distribution within the steps will be as shown in Fig. 64. There is ample practical evidence to show that this is in fact what occurs, except that the distribution is rarely as perfect as that shown (see B.S. 600R for typical examples). The

distribution of Fig. 64 can be shown as a curve (Fig. 65), familiar to those with a knowledge of statistical theory as the typical Gaussian curve.

According to pure theory, the ends of the curves approaching the zero line are asymptotic, meaning that the probability that a part may differ from the nominal size by a large amount is very small indeed but still positive, but in practice it is usual to consider that the curve ends when the probability of a part being outside the size corresponding to the extremity is 1 in 1000 (i.e. 2 in 1000

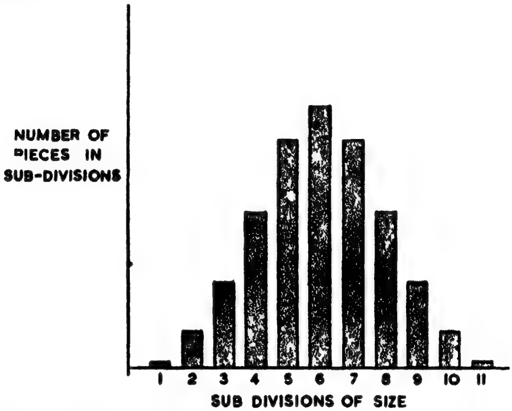


FIG 64 DIAGRAMMATIC REPRESENTATION OF DISTRIBUTION OF PARTS IN GRADED SIZES

considering both ends, or 99·8 per cent within limits) The figure shows clearly some remarkable facts—

99 per cent of components are within 83 per cent of the tolerance.
98 per cent of components are within 75·5 per cent of the tolerance.
95 per cent of components are within 63·5 per cent of the tolerance.
89 per cent of components are within 50 per cent of the tolerance.

This means that if the tolerances on a process were to be doubled, only 11 per cent of the parts would be outside the original limits, if the tolerance is increased by 50 per cent, only 4 per cent of the parts would be outside the original limits.

In practice, of course, other factors preclude advantage being taken of this to the full, but a method of making use of these facts will be described in a subsequent paragraph.

It is too optimistic on most machine processes to assume even or Gaussian distribution; usually the distribution is "skew," due to factors tending to make the part wrong in a particular direction; for example, roller box tool distortion on a capstan usually makes it turn oversize not undersize, and excessive speed by the operator will tend to produce large parts not small ones; hence a distribution skewed towards the upper limit is likely.

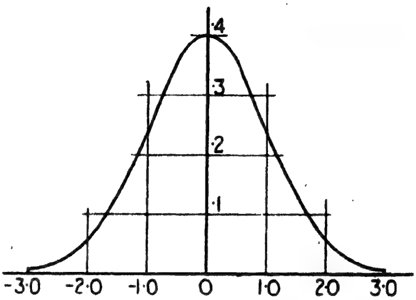


FIG. 65. THE GAUSSIAN CURVE

Fig. 66 shows the results of measuring 100 dowels drawn from a finished part stores. The work limits were 0·5 + 0·0005/+ 0 in., and inspection must have been 100 per cent, as no defectives were found on the recheck. The curve shows the distribution using steps of 0·0001 in. The abrupt drop in the curve on the

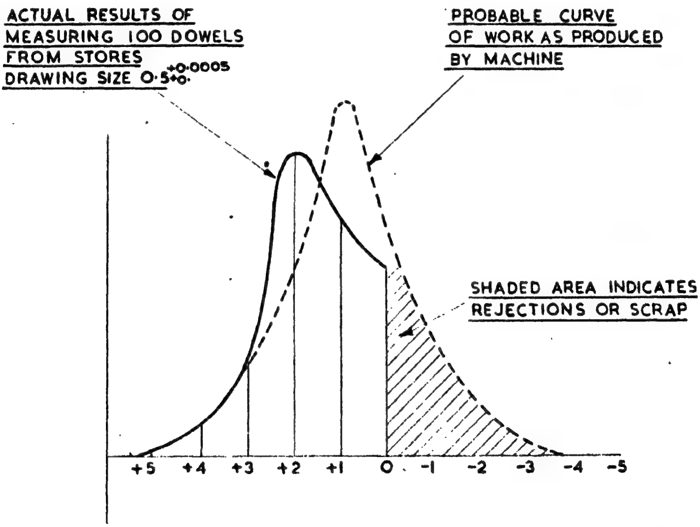


FIG. 66. THE RESULTS OF MEASURING 100 DOWELS

zero indicates (*a*) good initial inspection, and (*b*) that there must have been much initial scrap. The dotted line shows the probable Gaussian distribution, and the shift of the peak to the left and its reduction in height indicates that in all probability the inspection

was unduly cautious and rejected several parts which might have been accepted; alternatively, that the inspection gauge was reading about 0·0001 in. *low*. This illustration is interesting in that it shows what inferences on shop efficiency can be drawn from subsequent re-inspection of the shop's products. It also underlines the importance of the accuracy of inspection measurement devices in relation to the magnitude of the work tolerance.

When the machine itself is well capable of holding the tolerance given, as, for example, in centreless grinding a $\frac{1}{2}$ in. pin to a tolerance of 0·002 in., the usual practice of setting up at one extreme end of the tolerance and allowing the tool (grinding wheel, etc.) to wear gradually, results in Gaussian distribution as recorded on samples taken at regular intervals, but with the peak of the curve shifting gradually as the tool wears. In practice, what is done in the application of Quality Control is that at regular intervals samples are taken and measured, the size being recorded on a chart. The gradual shift of the mean can be seen from the general pattern of the chart.

4. Miscellaneous Definitions

There are other various expressions and methods used in the practical application of Quality Control which the designer should understand. For the full comprehension of the process there are numerous references which can be studied, but these are more likely to be used by the production engineer or inspector.

Deviation. The "deviation" of an observed result or measurement is its deviation from the average of all observations. The *standard deviation* is the square root of the mean of the squares of deviations of all the observations, and is thus equivalent to the familiar R.M.S. value in A.C. electrical engineering. The standard deviation (sigma, σ) of a Gaussian curve is shown on Fig. 65, and it is used as the basis for all control chart limits.

Sampling. Since 100 per cent inspection is rarely used on normal commercial quantity production, sampling inspection is used, and here Quality Control is particularly valuable, since statistical theory enables the quality of the whole (or "bulk") to be determined from the condition of the samples. The exact relation between quality in the sample and in the bulk (in terms of numbers of "defectives") has been determined exactly and the whole "inspection" process made into a means of warning the production man that trouble is impending, and not one of telling him after it has happened.

Defective. A defective is a part, not necessarily scrap, but outside some particular set of limits on a quality control chart or investigation.

5. The Use of the Theory of Probability in Widening Limits

Having once realized that in a properly controlled repetition manufacturing process something approaching Gaussian distribution of size will be achieved, and realizing also the significant facts mentioned above regarding the percentage of parts outside certain percentages of the tolerance, the designer should begin to wonder if he cannot widen his limits to make use of this knowledge.

At least one American ball-bearing manufacturer quotes bearing fits with a note, "In rare cases when the extremes of theoretical fits coincide a selection of bearings and shafts may be necessary." He quotes the "theoretical fit" as, say, + 0·0009, – 0·0001 in., and, separately, the "normal fit, 95 per cent of all cases," as + 0·0007, + 0·0001 in. From a total tolerance of 0·001 in. he has eliminated 0·0004 in. as affecting fits only 5 per cent of all cases.

This reasoning can be used in many simple cases where extremes of size may prevent correct assembly and where "selective rejection" can be made on assembly.

A common case is in the use of the size to size fits (i.e. U–L, H–h, etc.) where the hole lower limit and the shaft upper limit are both zero. The probability that both parts are the same size or even almost the same size is so small that this class of fit can be used as a light push or even close running fit. Again, however, if the limits are used on small or batch production it may give trouble, as it is quite easy to get skew distribution due to the operator influencing the size towards the zero. In many shops, however, this risk is accepted and occasional fitting allowed.

The true scientific process of determining limits taking into account probability effects can now be described. The case of a shaft fitting with controlled clearance (e.g. a bearing) will be considered. The steps are as follows—

(*a*) It must first be determined that the process to be used produces properly distributed (i.e. Gaussian) results.

(*b*) The designer sets by practical tests the minimum and maximum clearances, outside which performance will definitely suffer.

(*c*) The acceptable probability (P) that the clearance will be either inside or outside these limits is decided, taking into account such factors as the inconvenience such an occurrence would cause to the manufacturer or to the customer, and the ease with which matters could be put right. A probability of 1 in 1000 that the clearance is wrong is a likely one.

(*d*) This must be expressed in terms of probability of error on either shaft or hole (say, p_S, p_H).

$$P = p_S \times p_H$$

If $p_S = p_H$ and $P = 0{\cdot}001$, $p_S = p_H = \sqrt{0{\cdot}001} = 0{\cdot}0316$. In other words, it is likely that a 3 per cent risk of either shaft or hole being outside tolerance is acceptable.

(*e*) From published tables of properties of Gaussian curves (e.g. B.S. 600R, Table 9, "Gaussian Frequency Distribution") the size ordinates of the curve which give 97 per cent "good" results and $1\frac{1}{2}$ per cent rejects on either side can be determined. As explained previously, the Gaussian curve is arbitrarily chosen to end at

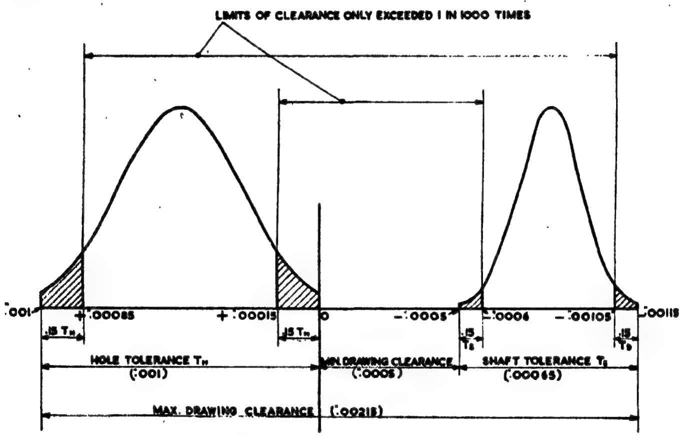


FIG. 67. FREQUENCY DISTRIBUTION CURVES FOR HOLE AND SHAFT—NORMAL METHOD OF SETTING LIMITS

99·8 per cent "good" results, corresponding to $\pm$ 3·09σ. The ordinates giving 97 per cent "good" results correspond approximately to $\pm$ 2·2σ.

(*f*) If the absolute theoretical tolerance on shaft and hole (assuming each is equal) is 100, the tolerance which would satisfy the probability risk accepted is therefore $\dfrac{100 \times 2{\cdot}2}{3{\cdot}09} = 71.$

(*g*) *The final tolerances which can be used are therefore* $\dfrac{100}{71}$, *or 40 per cent greater than would have been used were it not for allowing for the probability effect.*

As an example of the above process, the limits shown in Fig. 67 for a shaft and hole can be recalculated allowing for a probability of 1 in 1000 that the minimum (or maximum) clearance is exceeded. The revised limits and the method of deriving them are clearly shown in Fig. 68.

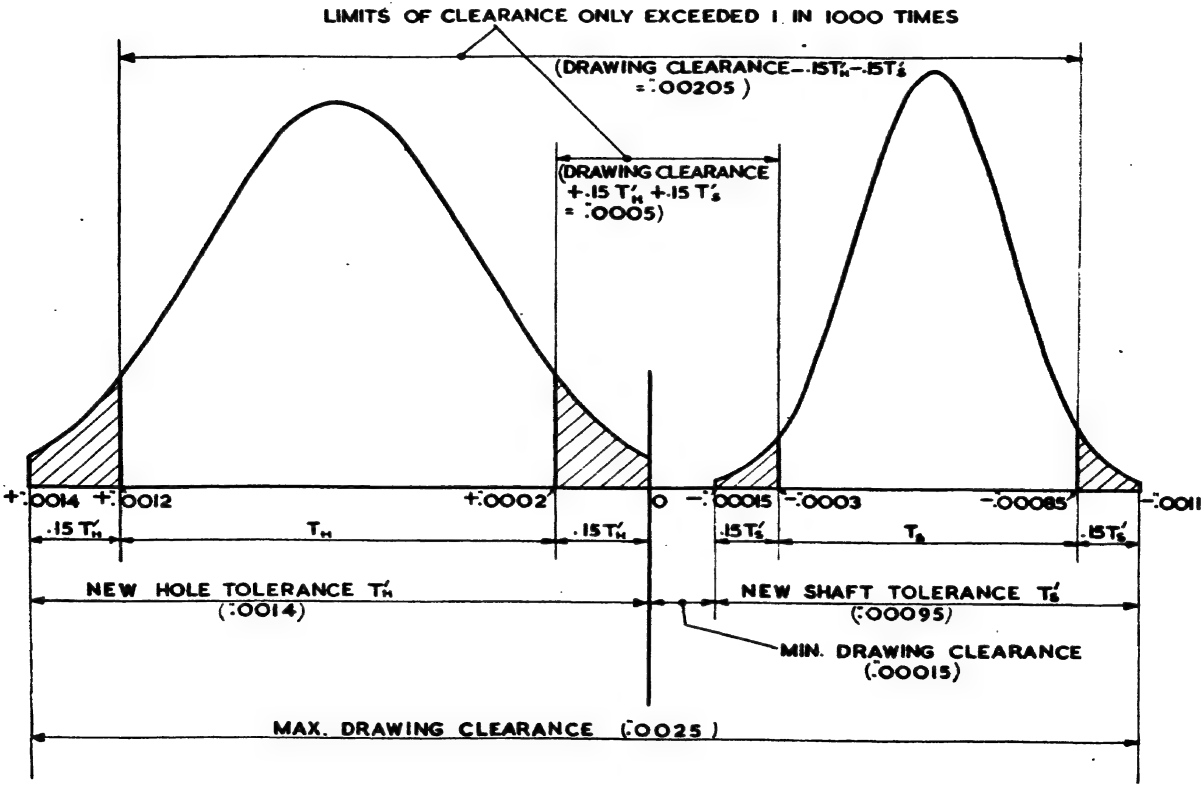


Fig. 68. Frequency Distribution Curves for Hole and Shaft—Limits set Allowing for Probability

In the original case the limits are—

Hole: + 0·001, + 0 (Tolerance 0·001)
Shaft: − 0·0005, − 0·00115 (Tolerance 0·00065)
Minimum clearance: 0·0005
Maximum clearance: 0·00215

In the revised case, the limits are—

Hole: + 0·0014, + 0 (Tolerance 0·0014)
Shaft: − 0·00015, − 0·0011 (Tolerance 0·00095)

The minimum clearance in 999 cases out of 1000 is still 0·0005. The maximum clearance in 999 cases out of 1000 is 0·00205.

6. Control Charts and their Interest to Designers

A Quality Control chart is primarily a production engineer's tool and will not concern the designer directly. Such charts are now widely used in this country and in the U.S.A., and without going into detail as to how the production engineer uses the chart to forecast and investigate, it is worth while considering how these charts can be used directly or indirectly to assist in the use of tolerances and limits.

In the first place, the fact that the chart can be prepared indicates that the process is a controlled one and that the tolerance set is a practicable one. The scatter of results over a short number of samples indicates the capabilities of the machine and the set-up in general, and thus enables the appropriate tolerance quality applicable to that type of machine to be determined. This information is of fundamental importance. Any gradual or otherwise drift of the mean of the results indicates tool wear or its equivalent and this can be considered in relation to the work tolerance. For example, it may be necessary because of difficulties with tool resetting to allow appreciable drift and thus a wide tolerance, subsequently compensated for by selective assembly; the chart will give precise information on the tool wear characteristics.

Control charts for several machines on one job will indicate the variations between different machines. Control charts for the same operation at different plants if correlated to service reports of performance can be used to trace defects or poor performance, and guide in the revision of limits. For example, if it is found that the best performance is with the most closely controlled production, it may mean that the extreme limits are not permitting proper performance (e.g. inadequate running clearance), although the mean values are perfectly satisfactory; the solution in this case is to decrease tolerances.

Finally, and by no means of least importance, control charts establish confidence between designer, production engineer, inspector and operator, and mitigate the condition where the designer uses tight tolerances, knowing that they will be exceeded; the production man widens them because he must, and the poor inspector is left to decide the appropriate degree of compromise.

PART II: STANDARD SYSTEMS

CHAPTER IV

STANDARD LIMIT AND TOLERANCE SYSTEMS

THE NEWALL TOLERANCE SYSTEM

THE Newall System has the distinction of being the oldest of surviving tolerance systems, and is also the simplest from the point of view of choice of fit. It is, however, considered by some to be inferior to other modern systems in certain respects. Its wide use now and in the past must be acknowledged.

It is listed in inches only.

1. Historical

The Newall Company has provided the following interesting notes on the history of the system—

"The compiler of the Tables was a Mr. John Walker Newall, who, during the latter part of the nineteenth century was engaged in the perfection of sheep-shearing machinery, and ultimately developed a type of machine for which there appeared to be prospects, and for the manufacture of which the Newall Company was established in 1900.

"The factory was situated in Featherstone Street, London, and it was there that the gauge business had its inception. A fair amount of trade was done with the Colonies in these sheep-shearing machines, particularly in Australia, and, in due course, orders followed for spare parts. Mr. Newall soon realized that, in view of the fact that the users were so far away, interchangeability of the parts was essential, and it was this that eventually led to the compilation of the Newall Tables, with the idea of ensuring such interchangeability.

"In those days very little had been done in this country in the matter of limits, although certain firms had drawn up tables to suit their own particular requirements. Mr. John Newall studied as many of these as were available, and also consulted a number of Continental and American firms who were working along the same lines.

"A great deal of information was gathered in this way, and it was the study, analysis and classification of this information that led to the compilation and publication of the Newall Standard Tables of Limits in the year 1902.

"The Newall system is founded on the 'hole' basis; that is, provision is made in the size of the hole for error in workmanship

only, variation to obtain the quality of fit required being allowed for on the size of the shaft which has to enter the hole. In almost all cases it is, of course, easier in practice to vary the size of the shaft than that of the hole, and it was, doubtless, this fact that prompted Mr. Newall to adopt this basis as against the shaft basis."

2. Grades of Fit

The system has two grades of bilateral holes (A and B), two grades of press fit (F and D), one transition fit (P), and three clearance fits (Z, Y, and X). These are shown diagrammatically in Fig. 69.

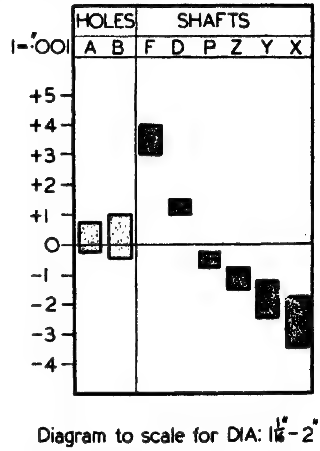


Fig. 69. Newall Limits Represented Diagrammatically

3. Association of Holes and Shafts

Either hole is intended to be associated with any shaft, but in practice the character of fit is correct in certain cases with a particular hole. The fit BD is not a true press fit below 1 in., although it is above; while PA is a true size and size fit, PB is a lower transition fit. The associations YA and XA are absurd.

4. Diameter Steps

The diameter steps are in poor progression and are $0-\frac{1}{2}$ in., $\frac{9}{16}-1$ in., $1\frac{1}{16}-2$ in., $2\frac{1}{16}-3$ in., $3\frac{1}{16}-4$ in., $4\frac{1}{16}-5$ in. These have been modified in Table 12 to eliminate the gap between the fractional sizes at the steps. What was intended above 5 in. is not known.

5. Derivation of Limits

The formulae from which the limits are supposed to be calculated are given below. The limits have been rounded off in a very empirical method and it is not clear which diameter D refers to, although it is probably the maximum diameter in the range, unless otherwise indicated.

A: $+ 0{\cdot}0006\sqrt{D}, \; - 0{\cdot}0003\sqrt{D}$

B: $+ 0{\cdot}0008\sqrt{D}, \; - 0{\cdot}0004\sqrt{D}$

F: Derived empirically: formulae approximately $+ 0{\cdot}003D^{\cdot 75}$, $+ 0{\cdot}0022D^{\cdot 87}$ where D is the geometric mean of the diameter steps.

D: Derived empirically: formulae approximately $+ 0{\cdot}0012D^{\cdot 75}$, $+ 0{\cdot}0008D^{\cdot 7}$ where D is the geometric mean of the diameter steps.

P: $- 0{\cdot}0002\sqrt{D}$, $- 0{\cdot}0006\sqrt{D}$

Z: $- 0{\cdot}0005\sqrt{D}$, $- 0{\cdot}001\sqrt{D}$

Y: $- 0{\cdot}001\sqrt{D}$, $- 0{\cdot}0018\sqrt{D}$

X: $- 0{\cdot}00125\sqrt{D}$, $- 0{\cdot}0025\sqrt{D}$

TABLE 12

NEWALL LIMITS (unit = 0·001 in.)

Diameter in.	Holes		Shafts					
	A	B	F	D	P	Z	Y	X
-0·5	+ 0·2 − 0·2	+ 0·5 − 0·5	+ 1 + 0·5	+ 0·5 + 0·2	− 0·2 − 0·7	− 0·5 − 0·7	− 0·7 − 1·2	− 1 − 2
0·501–1	+ 0·5 − 0·2	+ 0·7 − 0·5	+ 2 + 1·5	+ 1 + 0·7	− 0·2 − 0·7	− 0·7 − 1·2	− 1 − 2	− 1·2 − 2·7
1·001–2	+ 0·7 − 0·2	+ 1 − 0·5	+ 4 + 3	+ 1·5 + 1	− 0·2 − 0·7	− 0·7 − 1·5	− 1·2 − 2·5	− 1·7 − 3·5
2·001–3	+ 1 − 0·5	+ 1·2 − 0·7	+ 6 + 4·5	+ 2·5 + 1·5	− 0·5 − 1	− 1 − 2	− 1·5 − 3	− 2 − 4·2
3·001–4	+ 1 − 0·5	+ 1·5 − 0·7	+ 8 + 6	+ 3 + 2	− 0·5 − 1	− 1 − 2·2	− 2 − 3·5	− 2·5 − 5
4·001–5	+ 1 − 0·5	+ 1·7 − 0·7	+ 10 + 8	+ 3·5 + 2·5	− 0·5 − 1	− 1·2 − 2·5	− 2·2 − 4	− 3 − 5·7

Of interest particularly when comparing one system with another are the formulae for the more important *basic deviations* (D now being the G.M. of the diameter steps). Basic deviation of combination

$$BX = 0{\cdot}0009D^{\cdot 6}$$
$$BY = 0{\cdot}0006D^{\cdot 6}$$
$$BZ = 0{\cdot}00015D^{\cdot 7}$$
$$AZ = 0{\cdot}0004D^{\cdot 35}$$
$$AD = 0{\cdot}002D^{1\cdot 3}$$

As these have been derived from much rounded tables they are only approximate.

The relative magnitudes of the tolerances in the various fits are approximately in the following ratios—

A : B : P : X : Y : Z : F : D
1 : 3 : 1 : 3 : 2 : 1 : (special)

6. Description of Fits

As published by the Newall Company the various shaft fits have descriptions as follows, irrespective of which hole is used—

F—Force
D—Driving
P—Push
Z—Running (close)
Y—Running (normal)
X—Running (free)

7. Gauge Limits

No gauge limits applicable specifically to Newall limits are published. It is customary to use the equivalent B.S. 164 limits—i.e. those specified in B.S. 969. A tabulation of these is contained in Chapter VIII.

B.S. 164 LIMITS AND FITS FOR ENGINEERING

The current British Standard Tolerance System is that contained in B.S. 164 and is widely used. It is on the whole a satisfactory system, the chief criticism being that only one quality of shaft is listed in each fit and these are for many purposes too severe. A revised standard is under consideration by the B.S.I. at the moment of writing.

B.S. 164 lists inch and metric tolerances; the latter are unfortunately quoted in quarter units of 0·01 mm—i.e. 2·5 microns—which involves the use of ½ micron slip blocks, not normally obtainable. In the tables given in this book, this has been altered.

1. Historical

The original B.S. System was published in 1906 in B.S. 27, but this listed only three grades of running fit with associated holes. The holes were bilateral with the tolerance divided equally. Towards the end and after the war of 1914–18, when considerable progress had been made in quantity manufacture of munitions, various discussions and meetings were held, resulting in the issue of B.S. 164 in 1924. Some of the work carried out is recorded in the Thomas Hawksley Lecture at the Institution of Mechanical Engineers in 1920 by Sir R. T. Glazebrook—"Limit Gauging"—and also in an

interesting "Memorandum on Limits and Limit Gauges," prepared by Mr. R. Dumas and published by the B.S.I. in 1919.

2. Grades of Fit

The system has four recommended grades of hole (B, U, V, W) with unilateral limits, four equivalent grades of equally divided bilateral hole (K, X, Y, Z) and three oversize holes (A, G, H); an

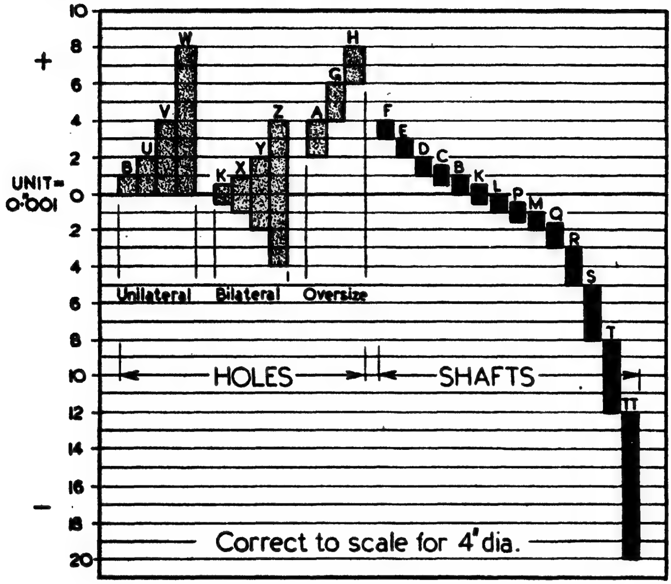


FIG. 70. B.S. 164 LIMITS REPRESENTED DIAGRAMMATICALLY

additional bilateral hole of very coarse tolerance (J) is also listed, but this is not intended for fits. Shafts are listed from reasonable interference to considerable clearance—(F, E, D, C, B, K, L, P, M, Q, R, S, T, and TT). L is the unilateral shaft with the upper limit zero. TT is not for fits.

These fits are shown diagrammatically in Fig. 70.

3. Association of Holes and Shafts

Any of the four qualities of hole both unilateral and bilateral are intended to be associated with any shaft, although the report recommends the use of U (or X bilateral) holes with the various shafts. The shaft tolerances correspond to the closest grade hole (B or K) and are comparatively tight (e.g. 0·0007 in. at 2 in. diameter, which corresponds to I.S.A. Quality 6). In the author's opinion this

is a bad feature, as it means that most of the shaft fits cannot be produced on a lathe but must be ground.

Owing to the lack of precise definition of the intended association, care must be taken with press fits that the fit is satisfactory with the hole chosen. The following gives the correct associations—

Press Fits	
Hole	Shaft
B, X	D, E, F
U, Y	E, F
K	C, D, E, F

4. Diameter Steps

The diameter steps are in a progression where the difference between successive steps (i.e. the amount of the step) is progressively $\frac{1}{10}$ in. (or 2·5 mm) greater. This is due to the particular parabolic type of curve from which the limits are derived. The full range of steps is as follows—

Inch: 0·3, 0·6, 1·0, 1·5, 2·1, 2·8, 3·6, 4·5, 5·5, 6·6, 7·8, 9·1, 10·5, 12·0, 13·6, 15·3, 17·1, 19·0, 21·0, 23·1, 25·3.

Metric: 7·5, 15, 25, 37·5, 52·5, 70, 90, 112·5, 137·5, 165, 195, 227·5, 262·5, 300, 340, 382·5, 427·5, 475, 525, 577·5, 632·5.

Diameters up to any size can be derived by extrapolation. A series of "size multipliers" are allocated to these steps, 3 being the step 0–0·29 in. (0–7·49 mm), 4 being the step 0·3–0·59 in. (7·5–14·99 mm), etc. These multipliers can be used for deriving the tolerances (see below).

5. Derivation of Limits

In the original specification the various fits are allocated "range factors", representing the limits at a hypothetical diameter (e.g. "E" = + 3/+ 2). When a range factor (deviation factor) is multiplied by the diameter size multipliers referred to above, the product is the limit of the fit in question. For example, the range factors for E are + 3/+ 2: multiplied by size multipliers for diameter 0·3–0·59 in. (= 4) gives the limits in thousandths of an inch (= + 1·2/+ 0·8 = 0·0012/0·0008 in.).

The original formula was of the type $0{\cdot}01\sqrt{20D} + 0{\cdot}01$ where D is the diameter step. However, for the sake of uniformity in comparing this system with others, the original formulae can be

TABLE 13.

LIMITS FOR HOLES: Inch Units

Nominal Sizes, in.	Unilateral Holes				
	B	U	V	W	K
0 to 0·29	+ 0·3	+ 0·6	+ 1·2	+ 2·4	+ 0·1
	+ 0	+ 0	+ 0	+ 0	− 0·2
0·3 „ 0·59	+ 0·4	+ 0·8	+ 1·6	+ 3·2	+ 0·2
	+ 0	+ 0	+ 0	+ 0	− 0·2
0·6 „ 0·99	+ 0·5	+ 1·0	+ 2·0	+ 4·0	+ 0·2
	+ 0	+ 0	+ 0	+ 0	− 0·3
1·0 „ 1·49	+ 0·6	+ 1·2	+ 2·4	+ 4·8	+ 0·3
	+ 0	+ 0	+ 0	+ 0	− 0·3
1·5 „ 2·09	+ 0·7	+ 1·4	+ 2·8	+ 5·6	+ 0·3
	+ 0	+ 0	+ 0	+ 0	− 0·4
2·1 „ 2·79	+ 0·8	+ 1·6	+ 3·2	+ 6·4	+ 0·4
	+ 0	+ 0	+ 0	+ 0	− 0·4
2·8 „ 3·59	+ 0·9	+ 1·8	+ 3·6	+ 7·2	+ 0·4
	+ 0	+ 0	+ 0	+ 0	− 0·5
3·6 „ 4·49	+ 1·0	+ 2·0	+ 4·0	+ 8·0	+ 0·5
	+ 0	+ 0	+ 0	+ 0	− 0·5
4·5 „ 5·49	+ 1·1	+ 2·2	+ 4·4	+ 8·8	+ 0·5
	+ 0	+ 0	+ 0	+ 0	− 0·6
5·5 „ 6·59	+ 1·2	+ 2·4	+ 4·8	+ 9·6	+ 0·6
	+ 0	+ 0	+ 0	+ 0	− 0·6
6·6 „ 7·79	+ 1·3	+ 2·6	+ 5·2	+ 10·4	+ 0·6
	+ 0	+ 0	+ 0	+ 0	− 0·7
7·8 „ 9·09	+ 1·4	+ 2·8	+ 5·6	+ 11·2	+ 0·7
	+ 0	+ 0	+ 0	+ 0	− 0·7
9·1 „ 10·49	+ 1·5	+ 3·0	+ 6·0	+ 12·0	+ 0·7
	+ 0	+ 0	+ 0	+ 0	− 0·8
10·5 „ 11·99	+ 1·6	+ 3·2	+ 6·4	+ 12·8	+ 0·8
	+ 0	+ 0	+ 0	+ 0	− 0·8
12·0 „ 13·59	+ 1·7	+ 3·4	+ 6·8	+ 13·6	+ 0·8
	+ 0	+ 0	+ 0	+ 0	− 0·9
13·6 „ 15·29	+ 1·8	+ 3·6	+ 7·2	+ 14·4	+ 0·9
	+ 0	+ 0	+ 0	+ 0	− 0·9
15·3 „ 17·09	+ 1·9	+ 3·8	+ 7·6	+ 15·2	+ 0·9
	+ 0	+ 0	+ 0	+ 0	− 1·0
17·1 „ 18·99	+ 2·0	+ 4·0	+ 8·0	+ 16·0	+ 1·0
	+ 0	+ 0	+ 0	+ 0	− 1·0
19·0 „ 20·99	+ 2·1	+ 4·2	+ 8·4	+ 16·8	+ 1·0
	+ 0	+ 0	+ 0	+ 0	− 1·1
21·0 „ 23·09	+ 2·2	+ 4·4	+ 8·8	+ 17·6	+ 1·1
	+ 0	+ 0	+ 0	+ 0	− 1·1
23·1 „ 25·29	+ 2·3	+ 4·6	+ 9·2	+ 18·4	+ 1·1
	+ 0	+ 0	+ 0	+ 0	− 1·2

B.S. 164 Limits
(Tolerance unit = 0·001 in.)

Bilateral Holes			Oversize Holes			Non-mating Holes and Shafts
X	Y	Z	A	G	H	J
+ 0·3	+ 0·6	+ 1·2	+ 1·2	+ 1·8	+ 2·4	+ 3·0
− 0·3	− 0·6	− 1·2	+ 0·6	+ 1·2	+ 1·8	− 3·0
+ 0·4	+ 0·8	+ 1·6	+ 1·6	+ 2·4	+ 3·2	+ 4·0
− 0·4	− 0·8	− 1·6	+ 0·8	+ 1·6	+ 2·4	− 4·0
+ 0·5	+ 1·0	+ 2·0	+ 2·0	+ 3·0	+ 4·0	+ 5·0
− 0·5	− 1·0	− 2·0	+ 1·0	+ 2·0	+ 3·0	− 5·0
+ 0·6	+ 1·2	+ 2·4	+ 2·4	+ 3·6	+ 4·8	+ 6·0
− 0·6	− 1·2	− 2·4	+ 1·2	+ 2·4	+ 3·6	− 6·0
+ 0·7	+ 1·4	+ 2·8	+ 2·8	+ 4·2	+ 5·6	+ 7·0
− 0·7	− 1·4	− 2·8	+ 1·4	+ 2·8	+ 4·2	− 7·0
+ 0·8	+ 1·6	+ 3·2	+ 3·2	+ 4·8	+ 6·4	+ 8·0
− 0·8	− 1·6	− 3·2	+ 1·6	+ 3·2	+ 4·8	− 8·0
+ 0·9	+ 1·8	+ 3·6	+ 3·6	+ 5·4	+ 7·2	+ 9·0
− 0·9	− 1·8	− 3·6	+ 1·8	+ 3·6	+ 5·4	− 9·0
+ 1·0	+ 2·0	+ 4·0	+ 4·0	+ 6·0	+ 8·0	+ 10·0
− 1·0	− 2·0	− 4·0	+ 2·0	+ 4·0	+ 6·0	− 10·0
+ 1·1	+ 2·2	+ 4·4	+ 4·4	+ 6·6	+ 8·8	+ 11·0
− 1·1	− 2·2	− 4·4	+ 2·2	+ 4·4	+ 6·6	− 11·0
+ 1·2	+ 2·4	+ 4·8	+ 4·8	+ 7·2	+ 9·6	+ 12·0
− 1·2	− 2·4	− 4·8	+ 2·4	+ 4·8	+ 7·2	− 12·0
+ 1·3	+ 2·6	+ 5·2	+ 5·2	+ 7·8	+ 10·4	+ 13·0
− 1·3	− 2·6	− 5·2	+ 2·6	+ 5·2	+ 7·8	− 13·0
+ 1·4	+ 2·8	+ 5·6	+ 5·6	+ 8·4	+ 11·2	+ 14·0
− 1·4	− 2·8	− 5·6	+ 2·8	+ 5·6	+ 8·4	− 14·0
+ 1·5	+ 3·0	+ 6·0	+ 6·0	+ 9·0	+ 12·0	+ 15·0
− 1·5	− 3·0	− 6·0	+ 3·0	+ 6·0	+ 9·0	− 15·0
+ 1·6	+ 3·2	+ 6·4	+ 6·4	+ 9·6	+ 12·8	+ 16·0
− 1·6	− 3·2	− 6·4	+ 3·2	+ 6·4	+ 9·6	− 16·0
+ 1·7	+ 3·4	+ 6·8	+ 6·8	+ 10·2	+ 13·6	+ 17·0
− 1·7	− 3·4	− 6·8	+ 3·4	+ 6·8	+ 10·2	− 17·0
+ 1·8	+ 3·6	+ 7·2	+ 7·2	+ 10·8	+ 14·4	+ 18·0
− 1·8	− 3·6	− 7·2	+ 3·6	+ 7·2	+ 10·8	− 18·0
+ 1·9	+ 3·8	+ 7·6	+ 7·6	+ 11·4	+ 15·2	+ 19·0
− 1·9	− 3·8	− 7·6	+ 3·8	+ 7·6	+ 11·4	− 19·0
+ 2·0	+ 4·0	+ 8·0	+ 8·0	+ 12·0	+ 16·0	+ 20·0
− 2·0	− 4·0	− 8·0	+ 4·0	+ 8·0	+ 12·0	− 20·0
+ 2·1	+ 4·2	+ 8·4	+ 8·4	+ 12·6	+ 16·8	+ 21·0
− 2·1	− 4·2	− 8·4	+ 4·2	+ 8·4	+ 12·6	− 21·0
+ 2·2	+ 4·4	+ 8·8	+ 8·8	+ 13·2	+ 17·6	+ 22·0
− 2·2	− 4·4	− 8·8	+ 4·4	+ 8·8	+ 13·2	− 22·0
+ 2·3	+ 4·6	+ 9·2	+ 9·2	+ 13·8	+ 18·4	+ 23·0
− 2·3	− 4·6	− 9·2	+ 4·6	+ 9·2	+ 13·8	− 23·0

TABLE

LIMITS FOR SHAFTS: Inch Units

Nominal Sizes, in.	F	E	D	C	B	K
0 to 0·29	+ 1·2 + 0·9	+ 0·9 + 0·6	+ 0·6 + 0·3	+ 0·4 + 0·1	+ 0·3 + 0	+ 0·1 − 0·2
0·3 „ 0·59	+ 1·6 + 1·2	+ 1·2 + 0·8	+ 0·8 + 0·4	+ 0·6 + 0·2	+ 0·4 + 0	+ 0·2 − 0·2
0·6 „ 0·99	+ 2·0 + 1·5	+ 1·5 + 1·0	+ 1·0 + 0·5	+ 0·7 + 0·2	+ 0·5 + 0	+ 0·2 − 0·3
1·0 „ 1·49	+ 2·4 + 1·8	+ 1·8 + 1·2	+ 1·2 + 0·6	+ 0·9 + 0·3	+ 0·6 + 0	+ 0·3 − 0·3
1·5 „ 2·09	+ 2·8 + 2·1	+ 2·1 + 1·4	+ 1·4 + 0·7	+ 1·0 + 0·3	+ 0·7 + 0	+ 0·3 − 0·4
2·1 „ 2·79	+ 3·2 + 2·4	+ 2·4 + 1·6	+ 1·6 + 0·8	+ 1·2 + 0·4	+ 0·8 + 0	+ 0·4 − 0·4
2·8 „ 3·59	+ 3·6 + 2·7	+ 2·7 + 1·8	+ 1·8 + 0·9	+ 1·3 + 0·4	+ 0·9 + 0	+ 0·4 − 0·5
3·6 „ 4·49	+ 4·0 + 3·0	+ 3·0 + 2·0	+ 2·0 + 1·0	+ 1·5 + 0·5	+ 1·0 + 0	+ 0·5 − 0·5
4·5 „ 5·49	+ 4·4 + 3·3	+ 3·3 + 2·2	+ 2·2 + 1·1	+ 1·6 + 0·5	+ 1·1 + 0	+ 0·5 − 0·6
5·5 „ 6·59	+ 4·8 + 3·6	+ 3·6 + 2·4	+ 2·4 + 1·2	+ 1·8 + 0·6	+ 1·2 + 0	+ 0·6 − 0·6
6·6 „ 7·79	+ 5·2 + 3·9	+ 3·9 + 2·6	+ 2·6 + 1·3	+ 1·9 + 0·6	+ 1·3 + 0	+ 0·6 − 0·7
7·8 „ 9·09	+ 5·6 + 4·2	+ 4·2 + 2·8	+ 2·8 + 1·4	+ 2·1 + 0·7	+ 1·4 + 0	+ 0·7 − 0·7
9·1 „ 10·49	+ 6·0 + 4·5	+ 4·5 + 3·0	+ 3·0 + 1·5	+ 2·2 + 0·7	+ 1·5 + 0	+ 0·7 − 0·8
10·5 „ 11·99	+ 6·4 + 4·8	+ 4·8 + 3·2	+ 3·2 + 1·6	+ 2·4 + 0·8	+ 1·6 + 0	+ 0·8 − 0·8
12·0 „ 13·59	+ 6·8 + 5·1	+ 5·1 + 3·4	+ 3·4 + 1·7	+ 2·5 + 0·8	+ 1·7 + 0	+ 0·8 − 0·9
13·6 „ 15·29	+ 7·2 + 5·4	+ 5·4 + 3·6	+ 3·6 + 1·8	+ 2·7 + 0·9	+ 1·8 + 0	+ 0·9 − 0·9
15·3 „ 17·09	+ 7·6 + 5·7	+ 5·7 + 3·8	+ 3·8 + 1·9	+ 2·8 + 0·9	+ 1·9 + 0	+ 0·9 − 1·0
17·1 „ 18·99	+ 8·0 + 6·0	+ 6·0 + 4·0	+ 4·0 + 2·0	+ 3·0 + 1·0	+ 2·0 + 0	+ 1·0 − 1·0
19·0 „ 20·99	+ 8·4 + 6·3	+ 6·3 + 4·2	+ 4·2 + 2·1	+ 3·1 + 1·0	+ 2·1 + 0	+ 1·0 − 1·1
21·0 „ 23·09	+ 8·8 + 6·6	+ 6·6 + 4·4	+ 4·4 + 2·2	+ 3·3 + 1·1	+ 2·2 + 0	+ 1·1 − 1·1
23·1 „ 25·29	+ 9·2 + 6·9	+ 6·9 + 4·6	+ 4·6 + 2·3	+ 3·4 + 1·1	+ 2·3 + 0	+ 1·1 − 1·2

13 (*contd.*)
(Tolerance unit = 0·001 in.)

L	P	M	Q	R	S	T	TT
− 0	− 0·2	− 0·3	− 0·5	− 0·9	− 1·5	− 2·4	− 3·6
− 0·3	− 0·5	− 0·6	− 0·9	− 1·5	− 2·4	− 3·6	− 6·0
− 0	− 0·2	− 0·4	− 0·6	− 1·2	− 2·0	− 3·2	− 4·8
− 0·4	− 0·6	− 0·8	− 1·2	− 2·0	− 3·2	− 4·8	− 8·0
− 0	− 0·3	− 0·5	− 0·8	− 1·5	− 2·5	− 4·0	− 6·0
− 0·5	− 0·8	− 1·0	− 1·5	− 2·5	− 4·0	− 6·0	− 10·0
− 0	− 0·3	− 0·6	− 0·9	− 1·8	− 3·0	− 4·8	− 7·2
− 0·6	− 0·9	− 1·2	− 1·8	− 3·0	− 4·8	− 7·2	− 12·0
− 0	− 0·4	− 0·7	− 1·1	− 2·1	− 3·5	− 5·6	− 8·4
− 0·7	− 1·1	− 1·4	− 2·1	− 3·5	− 5·6	− 8·4	− 14·0
− 0	− 0·4	− 0·8	− 1·2	− 2·4	− 4·0	− 6·4	− 9·6
− 0·8	− 1·2	− 1·6	− 2·4	− 4·0	− 6·4	− 9·6	− 16·0
− 0	− 0·5	− 0·9	− 1·4	− 2·7	− 4·5	− 7·2	− 10·8
− 0·9	− 1·4	− 1·8	− 2·7	− 4·5	− 7·2	− 10·8	− 18·0
− 0	− 0·5	− 1·0	− 1·5	− 3·0	− 5·0	− 8·0	− 12·0
− 1·0	− 1·5	− 2·0	− 3·0	− 5·0	− 8·0	− 12·0	− 20·0
− 0	− 0·6	− 1·1	− 1·7	− 3·3	− 5·5	− 8·8	− 13·2
− 1·1	− 1·7	− 2·2	− 3·3	− 5·5	− 8·8	− 13·2	− 22·0
− 0	− 0·6	− 1·2	− 1·8	− 3·6	− 6·0	− 9·6	− 14·4
− 1·2	− 1·8	− 2·4	− 3·6	− 6·0	− 9·6	− 14·4	− 24·0
− 0	− 0·7	− 1·3	− 2·0	− 3·9	− 6·5	− 10·4	− 15·6
− 1·3	− 2·0	− 2·6	− 3·9	− 6·5	− 10·4	− 15·6	− 26·0
− 0	− 0·7	− 1·4	− 2·1	− 4·2	− 7·0	− 11·2	− 16·8
− 1·4	− 2·1	− 2·8	− 4·2	− 7·0	− 11·2	− 16·8	− 28·0
− 0	− 0·8	− 1·5	− 2·3	− 4·5	− 7·5	− 12·0	− 18·0
− 1·5	− 2·3	− 3·0	− 4·5	− 7·5	− 12·0	− 18·0	− 30·0
− 0	− 0·8	− 1·6	− 2·4	− 4·8	− 8·0	− 12·8	− 19·2
− 1·6	− 2·4	− 3·2	− 4·8	− 8·0	− 12·8	− 19·2	− 32·0
− 0	− 0·9	− 1·7	− 2·6	− 5·1	− 8·5	− 13·6	− 20·4
− 1·7	− 2·6	− 3·4	− 5·1	− 8·5	− 13·6	− 20·4	− 34·0
− 0	− 0·9	− 1·8	− 2·7	− 5·4	− 9·0	− 14·4	− 21·6
− 1·8	− 2·7	− 3·6	− 5·4	− 9·0	− 14·4	− 21·6	− 36·0
− 0	− 1·0	− 1·9	− 2·9	− 5·7	− 9·5	− 15·2	− 22·8
− 1·9	− 2·9	− 3·8	− 5·7	− 9·5	− 15·2	− 22·8	− 38·0
− 0	− 1·0	− 2·0	− 3·0	− 6·0	− 10·0	− 16·0	− 24·0
− 2·0	− 3·0	− 4·0	− 6·0	− 10·0	− 16·0	− 24·0	− 40·0
− 0	− 1·1	− 2·1	− 3·2	− 6·3	− 10·5	− 16·8	− 25·2
− 2·1	− 3·2	− 4·2	− 6·3	− 10·5	− 16·8	− 25·2	− 42·0
− 0	− 1·1	− 2·2	− 3·3	− 6·6	− 11·0	− 17·6	− 26·4
− 2·2	− 3·3	− 4·4	− 6·6	− 11·0	− 17·6	− 26·4	− 44·0
− 0	− 1·2	− 2·3	− 3·5	− 6·9	− 11·5	− 18·4	− 27·6
− 2·3	− 3·5	− 4·6	− 6·9	− 11·5	− 18·4	− 27·6	− 46·0

TABLE

LIMITS FOR HOLES: Metric Units

Nominal Sizes, mm	Unilateral Holes				
	B	U	V	W	K
0 to 7·49	+ 7	+ 15	+ 30	+ 60	+ 2
	+ 0	+ 0	+ 0	+ 0	− 5
7·5 ,, 14·99	+ 10	+ 20	+ 40	+ 80	+ 5
	+ 0	+ 0	+ 0	+ 0	− 5
15·0 ,, 24·99	+ 12	+ 25	+ 50	+ 100	+ 5
	+ 0	+ 0	+ 0	+ 0	− 7
25·0 ,, 37·49	+ 15	+ 30	+ 60	+ 120	+ 7
	+ 0	+ 0	+ 0	+ 0	− 7
37·5 ,, 52·49	+ 17	+ 35	+ 70	+ 140	+ 7
	+ 0	+ 0	+ 0	+ 0	− 10
52·5 ,, 69·99	+ 20	+ 40	+ 80	+ 160	+ 10
	+ 0	+ 0	+ 0	+ 0	− 10
70·0 ,, 89·99	+ 22	+ 45	+ 90	+ 180	+ 10
	+ 0	+ 0	+ 0	+ 0	− 12
90·0 ,, 112·49	+ 25	+ 50	+ 100	+ 200	+ 12
	+ 0	+ 0	+ 0	+ 0	− 12
112·5 ,, 137·49	+ 27	+ 55	+ 110	+ 220	+ 12
	+ 0	+ 0	+ 0	+ 0	− 15
137·5 ,, 164·99	+ 30	+ 60	+ 120	+ 240	+ 15
	+ 0	+ 0	+ 0	+ 0	− 15
165·0 ,, 194·99	+ 32	+ 65	+ 130	+ 260	+ 15
	+ 0	+ 0	+ 0	+ 0	− 17
195·0 ,, 227·49	+ 35	+ 70	+ 140	+ 280	+ 17
	+ 0	+ 0	+ 0	+ 0	− 17
227·5 ,, 262·49	+ 37	+ 75	+ 150	+ 300	+ 17
	+ 0	+ 0	+ 0	+ 0	− 20
262·5 ,, 299·99	+ 40	+ 80	+ 160	+ 320	+ 20
	+ 0	+ 0	+ 0	+ 0	− 20
300·0 ,, 339·99	+ 42	+ 85	+ 170	+ 340	+ 20
	+ 0	+ 0	+ 0	+ 0	− 22
340·0 ,, 382·49	+ 45	+ 90	+ 180	+ 360	+ 22
	+ 0	+ 0	+ 0	+ 0	− 22
382·5 ,, 427·49	+ 47	+ 95	+ 190	+ 380	+ 22
	+ 0	+ 0	+ 0	+ 0	− 25
427·5 ,, 474·99	+ 50	+ 100	+ 200	+ 400	+ 25
	+ 0	+ 0	+ 0	+ 0	− 25
475·0 ,, 524·99	+ 52	+ 105	+ 210	+ 420	+ 25
	+ 0	+ 0	+ 0	+ 0	− 27
525·0 ,, 577·49	+ 55	+ 110	+ 220	+ 440	+ 27
	+ 0	+ 0	+ 0	+ 0	− 27
577·5 ,, 632·49	+ 57	+ 115	+ 230	+ 460	+ 27
	+ 0	+ 0	+ 0	+ 0	− 30

13 (*contd.*)
(Tolerance unit = 0·001 mm)

Bilateral Holes			Oversize Holes			Non-mating Holes and Shafts
X	Y	Z	A	G	H	J
+ 7	+ 15	+ 30	+ 30	+ 45	+ 60	+ 60
− 7	− 15	− 30	+ 15	+ 30	+ 45	− 60
+ 10	+ 20	+ 40	+ 40	+ 60	+ 80	+ 80
− 10	− 20	− 40	+ 20	+ 40	+ 60	− 80
+ 12	+ 25	+ 50	+ 50	+ 75	+ 100	+ 100
− 12	− 25	− 50	+ 25	+ 50	+ 75	− 100
+ 15	+ 30	+ 60	+ 60	+ 90	+ 120	+ 120
− 15	− 30	− 60	+ 30	+ 60	+ 90	− 120
+ 17	+ 35	+ 70	+ 70	+ 105	+ 140	+ 140
− 17	− 35	− 70	+ 35	+ 70	+ 105	− 140
+ 20	+ 40	+ 80	+ 80	+ 120	+ 160	+ 160
− 20	− 40	− 80	+ 40	+ 80	+ 120	− 160
+ 22	+ 45	+ 90	+ 90	+ 135	+ 180	+ 180
− 22	− 45	− 90	+ 45	+ 90	+ 135	− 180
+ 25	+ 50	+ 100	+ 100	+ 150	+ 200	+ 200
− 25	− 50	− 100	+ 50	+ 100	+ 150	− 200
+ 27	+ 55	+ 110	+ 110	+ 165	+ 220	+ 220
− 27	− 55	− 110	+ 55	+ 110	+ 165	− 220
+ 30	+ 60	+ 120	+ 120	+ 180	+ 240	+ 240
− 30	− 60	− 120	+ 60	+ 120	+ 180	− 240
+ 32	+ 65	+ 130	+ 130	+ 195	+ 260	+ 260
− 32	− 65	− 130	+ 65	+ 130	+ 195	− 260
+ 35	+ 70	+ 140	+ 140	+ 210	+ 280	+ 280
− 35	− 70	− 140	+ 70	+ 140	+ 210	− 280
+ 37	+ 75	+ 150	+ 150	+ 225	+ 300	+ 300
− 37	− 75	− 150	+ 75	+ 150	+ 225	− 300
+ 40	+ 80	+ 160	+ 160	+ 240	+ 320	+ 320
− 40	− 80	− 160	+ 80	+ 160	+ 240	− 320
+ 42	+ 85	+ 170	+ 170	+ 255	+ 340	+ 340
− 42	− 85	− 170	+ 85	+ 170	+ 255	− 340
+ 45	+ 90	+ 180	+ 180	+ 270	+ 360	+ 360
− 45	− 90	− 180	+ 90	+ 180	+ 270	− 360
+ 47	+ 95	+ 190	+ 190	+ 285	+ 380	+ 380
− 47	− 95	− 190	+ 95	+ 190	+ 285	− 380
+ 50	+ 100	+ 200	+ 200	+ 300	+ 400	+ 400
− 50	− 100	− 200	+ 100	+ 200	+ 300	− 400
+ 52	+ 105	+ 210	+ 210	+ 315	+ 420	+ 420
− 52	− 105	− 210	+ 105	+ 210	+ 315	− 420
+ 55	+ 110	+ 220	+ 220	+ 330	+ 440	+ 440
− 55	− 110	− 220	+ 110	+ 220	+ 330	− 440
+ 57	+ 115	+ 230	+ 230	+ 345	+ 460	+ 460
− 57	− 115	− 230	+ 115	+ 230	+ 345	− 460

TABLE

LIMITS FOR SHAFTS: Metric Unit

Nominal Sizes, mm	F	E	D	C	B	K
0 to 7·49	+ 30	+ 22	+ 15	+ 10	+ 7	+ 2
	+ 22	+ 15	+ 7	+ 2	+ 0	− 5
7·5 „ 14·99	+ 40	+ 30	+ 20	+ 15	+ 10	+ 5
	+ 30	+ 20	+ 10	+ 5	+ 0	− 5
15·0 „ 24·99	+ 50	+ 37	+ 25	+ 17	+ 12	+ 5
	+ 37	+ 25	+ 12	+ 5	+ 0	− 7
25·0 „ 37·49	+ 60	+ 45	+ 30	+ 22	+ 15	+ 7
	+ 45	+ 30	+ 15	+ 7	+ 0	− 7
37·5 „ 52·49	+ 70	+ 52	+ 35	+ 25	+ 17	+ 7
	+ 52	+ 35	+ 17	+ 7	+ 0	− 10
52·5 „ 69·99	+ 80	+ 60	+ 40	+ 30	+ 20	+ 10
	+ 60	+ 40	+ 20	+ 10	+ 0	− 10
70·0 „ 89·99	+ 90	+ 67	+ 45	+ 32	+ 22	+ 10
	+ 67	+ 45	+ 22	+ 10	+ 0	− 12
90·0 „ 112·49	+ 100	+ 75	+ 50	+ 37	+ 25	+ 12
	+ 75	+ 50	+ 25	+ 12	+ 0	− 12
112·5 „ 137·49	+ 110	+ 82	+ 55	+ 40	+ 27	+ 12
	+ 82	+ 55	+ 27	+ 12	+ 0	− 15
137·5 „ 164·99	+ 120	+ 90	+ 60	+ 45	+ 30	+ 15
	+ 90	+ 60	+ 30	+ 15	+ 0	− 15
165·0 „ 194·99	+ 130	+ 97	+ 65	+ 47	+ 32	+ 15
	+ 97	+ 65	+ 32	+ 15	+ 0	− 17
195·0 „ 227·49	+ 140	+ 105	+ 70	+ 52	+ 35	+ 17
	+ 105	+ 70	+ 35	+ 17	+ 0	− 17
227·5 „ 262·49	+ 150	+ 112	+ 75	+ 55	+ 37	+ 17
	+ 112	+ 75	+ 37	+ 17	+ 0	− 20
262·5 „ 299·99	+ 160	+ 120	+ 80	+ 60	+ 40	+ 20
	+ 120	+ 80	+ 40	+ 20	+ 0	− 20
300·0 „ 339·99	+ 170	+ 127	+ 85	+ 62	+ 42	+ 20
	+ 127	+ 85	+ 42	+ 20	+ 0	− 22
340·0 „ 382·49	+ 180	+ 135	+ 90	+ 67	+ 45	+ 22
	+ 135	+ 90	+ 45	+ 22	+ 0	− 22
382·5 „ 427·49	+ 190	+ 142	+ 95	+ 70	+ 47	+ 22
	+ 142	+ 95	+ 47	+ 22	+ 0	− 25
427·5 „ 474·99	+ 200	+ 150	+ 100	+ 75	+ 50	+ 25
	+ 150	+ 100	+ 50	+ 25	+ 0	− 25
475·0 „ 524·99	+ 210	+ 157	+ 105	+ 77	+ 52	+ 25
	+ 157	+ 105	+ 52	+ 25	+ 0	− 27
525·0 „ 577·49	+ 220	+ 165	+ 110	+ 82	+ 55	+ 27
	+ 165	+ 110	+ 55	+ 27	+ 0	− 27
577·5 „ 632·49	+ 230	+ 172	+ 115	+ 85	+ 57	+ 27
	+ 172	+ 115	+ 57	+ 27	+ 0	− 30

13 (*contd.*)
(Tolerance unit = 0·001 mm)

L	P	M	Q	R	S	T	TT
− 0	− 5	− 7	− 12	− 22	− 37	− 60	− 90
− 7	− 12	− 15	− 22	− 37	− 60	− 90	− 150
− 0	− 5	− 10	− 15	− 30	− 50	− 80	− 120
− 10	− 15	− 20	− 30	− 50	− 80	− 120	− 200
− 0	− 7	− 12	− 20	− 37	− 62	− 100	− 150
− 12	− 20	− 25	− 37	− 62	− 100	− 150	− 250
− 0	− 7	− 15	− 25	− 45	− 75	− 120	− 180
− 15	− 22	− 30	− 45	− 75	− 120	− 180	− 300
− 0	− 10	− 17	− 27	− 52	− 87	− 140	− 210
− 17	− 27	− 35	− 52	− 87	− 140	− 210	− 350
− 0	− 10	− 20	− 30	− 60	− 100	− 160	− 240
− 20	− 30	− 40	− 60	− 100	− 160	− 240	− 400
− 0	− 12	− 22	− 35	− 67	− 112	− 180	− 270
− 22	− 35	− 45	− 67	− 112	− 180	− 270	− 450
− 0	− 12	− 25	− 37	− 75	− 125	− 200	− 300
− 25	− 37	− 50	− 75	− 125	− 200	− 300	− 500
− 0	− 15	− 27	− 42	− 82	− 137	− 220	− 330
− 27	− 42	− 55	− 82	− 137	− 220	− 330	− 550
− 0	− 15	− 30	− 45	− 90	− 150	− 240	− 360
− 30	− 45	− 60	− 90	− 150	− 240	− 360	− 600
− 0	− 17	− 32	− 50	− 97	− 162	− 260	− 390
− 32	− 50	− 65	− 97	− 162	− 260	− 390	− 650
− 0	− 17	− 35	− 52	− 105	− 175	− 280	− 420
− 35	− 52	− 70	− 105	− 175	− 280	− 420	− 700
− 0	− 20	− 37	− 57	− 112	− 187	− 300	− 450
− 37	− 57	− 75	− 112	− 187	− 300	− 450	− 750
− 0	− 20	− 40	− 60	− 120	− 200	− 320	− 480
− 40	− 60	− 80	− 120	− 200	− 320	− 480	− 800
− 0	− 22	− 42	− 65	− 127	− 212	− 340	− 510
− 42	− 65	− 85	− 127	− 212	− 340	− 510	− 850
− 0	− 22	− 45	− 67	− 135	− 225	− 360	− 540
− 45	− 67	− 90	− 135	− 225	− 360	− 540	− 900
− 0	− 25	− 47	− 72	− 142	− 237	− 380	− 570
− 47	− 72	− 95	− 142	− 237	− 380	− 570	− 950
− 0	− 25	− 50	− 75	− 150	− 250	− 400	− 600
− 50	− 75	− 100	− 150	− 250	− 400	− 600	− 1000
− 0	− 27	− 52	− 80	− 157	− 262	− 420	− 630
− 52	− 80	− 105	− 157	− 262	− 420	− 630	− 1050
− 0	− 27	− 55	− 82	− 165	− 275	− 440	− 660
− 55	− 82	− 110	− 165	− 275	− 440	− 660	− 1100
− 0	− 30	− 57	− 87	− 172	− 287	− 460	− 690
− 57	− 87	− 115	− 172	− 287	− 460	− 690	− 1150

converted with good accuracy (2 per cent) to the formulae shown in Table 14 based on multiples of $D^{\cdot 445}$ where D is the geometric mean of the diameter steps in use.

TABLE 14

DERIVATION OF BASIC DEVIATIONS OR MORE IMPORTANT LIMITS

Designation	Limit	Formula (unit = 0·001 in.)	Ratio
Shaft F	Lower	$+ 0{\cdot}00165D^{0{\cdot}445}$	6
" E	"	$+ 0{\cdot}00110D^{0{\cdot}445}$	4
" D	"	$+ 0{\cdot}00055D^{0{\cdot}445}$	2
" C	"	$+ 0{\cdot}000275D^{0{\cdot}445}$	1
" B	"	0	0
" L	Upper	0	0
" P	"	$- 0{\cdot}000275D^{0{\cdot}445}$	1
" M	"	$- 0{\cdot}00055D^{0{\cdot}445}$	2
" Q	"	$- 0{\cdot}000825D^{0{\cdot}445}$	3
" R	"	$- 0{\cdot}00165D^{0{\cdot}445}$	6
" S	"	$- 0{\cdot}00275D^{0{\cdot}445}$	10
" T	"	$- 0{\cdot}0044D^{0{\cdot}445}$	16
" TT	"	$- 0{\cdot}0066D^{0{\cdot}445}$	24
Oversize hole A	Lower	$+ 0{\cdot}0011D^{0{\cdot}445}$	4
" " G	"	$+ 0{\cdot}0022D^{0{\cdot}445}$	8
" " H	"	$+ 0{\cdot}0033D^{0{\cdot}445}$	12

In the case of shaft K, J, and holes K, X, Y, Z, and J the tolerance is divided equally about the zero except that an odd number is divided with the greater number negative—e.g. 1·5 gives + 0·7, − 0·8.

It will be noted that the formulae are all of the same form; interference fits should properly be proportional to the diameter, and therefore the above interference fits have to be determined for each diameter *ab initio*.

TABLE 15

DERIVATION AND RELATIVE MAGNITUDES OF THE TOLERANCES

Designation	Formula (unit = 0·001 in.)	Ratio
Holes B, K	$0{\cdot}00055D^{0{\cdot}445}$	1
" U, X, A, G, H	$0{\cdot}0011D^{0{\cdot}445}$	2
" V, Y	$0{\cdot}0022D^{0{\cdot}445}$	4
" W, Z	$0{\cdot}0044D^{0{\cdot}445}$	8
Hole J	$0{\cdot}011D^{0{\cdot}445}$	20
Shafts F, E, D, C, B, K, L, P, M	$0{\cdot}00055D^{0{\cdot}445}$	1
Shaft Q	$0{\cdot}000825D^{0{\cdot}445}$	1·5
" R	$0{\cdot}0011D^{0{\cdot}445}$	2
" S	$0{\cdot}00165D^{0{\cdot}445}$	3
" T	$0{\cdot}0022D^{0{\cdot}445}$	4
" TT	$0{\cdot}0044D^{0{\cdot}445}$	8

The formulae shown in Table 15 give the magnitude of the tolerances themselves, D again being the geometric mean of the diameter steps.

The "Quality" or fundamental tolerance series to be compared with the I.S.A. Standard Number series is therefore

1, 1·5, 2, 3, 4, 8, 20

6. Description of Fits

The standard does not recommend the use of descriptive titles for the various fits, but lists the following as a guide—

Class	Description	Unilateral	Bilateral
Interference	Force	—	X F
	Heavy Drive	U F	X E
	Light Drive	U E	X D
Transition	Extra Light Drive	—	X C
	Heavy Keying	U D	X B
	Medium Keying	U C	X K
	Light Keying	U B	X L
	Push	U K	X P
Clearance	Slide or Easy Push	U L	X M
	Easy Slide or Close Running	U P	X Q
	Close Running (1)	U M	—
	Close Running (2)	U Q	—
	Normal Running	U R	X R
	Slack Running	U S	X S
	Extra Slack Running	U T	X T
	Coarse Clearance	U TT	X TT

7. Gauge Limits

No gauge limits applicable to these limits were published until 1941 in B.S. 969. These are in abbreviated form applicable to any tolerance system, inch or metric, and are discussed and tabulated in Chapter VIII.

THE AMERICAN STANDARDS ASSOCIATION TOLERANCE SYSTEM

The only current American tolerance system is that contained in A.S.A. Standard B4a, published in 1925.* This is an early attempt

* A revision B4.1–1947 (Part I) was published in 1947, but contained only references to terminology, preferred sizes and a range of standardized tolerance units to be applied to all tolerances and allowances (deviations). No actual limits or fits are listed. The range of tolerance units is: (unit 0·001 in.). 0·1, 0·15, 0·2, 0·25, 0·3, 0·4, 0·5, 0·6, 0·8, 1, 1·2, 1·5, 2·0, 2·5, 3, 4, 5, 6, 8, 10, 12, 15, 20, 25, 30. Part II, containing the tables of fits, is to be published in due course.

at a "compromise" system using the knowledge available at the time of issue and taking into account the conflicting requirements of different production designs. The solution reached enabled tables to be published covering a few common fits and left individual designers to create the many other fits required in practice. Selective assembly was considered necessary with many of the fits and the standard is unique in this respect.

It is known that the standard is not considered satisfactory in the U.S.A. and is under revision to broaden its scope.

1. Grades of Fit

The system lists eight fits, each shaft having one appropriate hole only, although fine fits use the same hole. The holes are unilateral. The fits are described only and have no symbol reference. The descriptions are as follows—

	Description	
1.	Heavy Force or Shrink fit	with Hole (1)
2.	Medium Force fit	with Hole (1)
3.	Tight fit	with Hole (1)
4.	Wringing fit	with Hole (1)
5.	Snug fit	with Hole (1)
6.	Medium fit	with Hole (2)
7.	Free fit	with Hole (3)
8.	Loose fit	with Hole (4)

The fits are shown diagrammatically in Fig. 71.

2. Association of Holes and Shafts

As already indicated, the associations are given and fixed. However, the standard indicates that only the last four fits (Loose–Snug) are intended for interchangeable manufacture and that it is intended that selective assembly shall be used with the remainder to achieve the desired fits.

3. Diameter Steps

The diameter steps are in a very rough geometric progression based on fractional sizes. The arithmetic mean of the extremities of a step is a common fraction, as will be seen from the list of steps, as follows—

$0, \frac{3}{16}, \frac{5}{16}, \frac{7}{16}, \frac{9}{16}, \frac{11}{16}, \frac{13}{16}, \frac{15}{16}, 1\frac{1}{16}, 1\frac{3}{16}, 1\frac{3}{8}, 1\frac{5}{8}, 1\frac{7}{8}, 2\frac{1}{8}, 2\frac{3}{8}, 2\frac{3}{4}, 3\frac{1}{4}, 3\frac{3}{4}, 4\frac{1}{4}, 4\frac{3}{4}, 5\frac{1}{2}, 6\frac{1}{2}, 7\frac{1}{2}, 8\frac{1}{2}$.

Above this diameter only the two press fits (Medium Force and Heavy Force) are listed, but they continue in steps—

$9\frac{1}{2}$, $11\frac{1}{2}$, 13, 15, 17, 19, 22, 26, 30, 34, 38, 44, 52, 60, 68, 76, 88, 104, 120, 136.

(There is, incidentally, a mistake in the original standard on both tables, and for "From $10\frac{1}{2}$ in." read "From $9\frac{1}{2}$ in.").

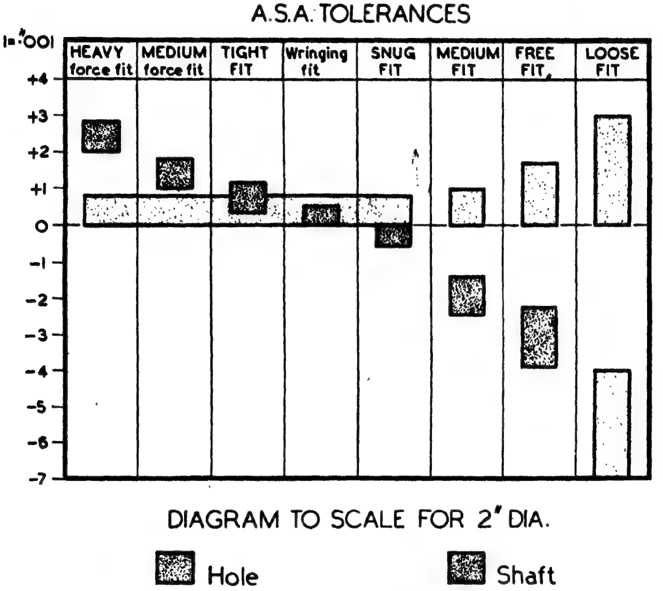


Fig. 71. A.S.A. Limits Represented Diagrammatically

Owing to rounding off, many of the steps are joined together in some of the tables.

4. Derivation of Limits

The formulae for calculating all limits are given in the standard and are listed in Table 17. The limits are rounded off to the nearest 0·0001 in., and the diameter D is the arithmetic mean of the diameter step extremities (except in one or two cases, e.g. $9\frac{1}{2}$–$11\frac{1}{2}$, "mean" = 10; $11\frac{1}{2}$–13, "mean" = 12). Owing to the smallness of the steps the errors in using the arithmetic as opposed to the geometric mean of the diameter steps, as in other parts of this chapter, are negligible.

TABLE 16

A.S.A. Limits

(Tolerance unit = 0·001 in.)

Diameter, in.	Holes 1	Holes 2	Holes 3	Holes 4	Shafts: With Hole 1 Heavy Force	With Hole 1 Medium Force	With Hole 1 Tight	With Hole 1 Wringing	With Hole 1 Snug	With Hole 2 Medium	With Hole 3 Free	With Hole 4 Loose
– 0·1875	+ 0·3	+ 0·4	+ 0·7	+ 1·0	+ 0·4	+ 0·4	+ 0·3	+ 0·2	– 0	– 0·2	– 0·4	– 1·0
	+ 0	+ 0	+ 0	+ 0	+ 0·1	+ 0·1	+ 0	+ 0	– 0·2	– 0·6	– 1·1	– 2·0
0·188– 0·3125	+ 0·4	+ 0·5	+ 0·8	+ 2·0	+ 0·7	+ 0·5	+ 0·5	+ 0·3	– 0	– 0·4	– 0·6	– 1·0
	+ 0	+ 0	+ 0	+ 0	+ 0·3	+ 0·1	+ 0·1	+ 0	– 0·3	– 0·9	– 1·4	– 3·0
0·313– 0·4375	+ 0·4	+ 0·6	+ 0·9	+ 2·0	+ 0·8	+ 0·6	+ 0·5	+ 0·3	– 0	– 0·5	– 0·7	– 1·0
	+ 0	+ 0	+ 0	+ 0	+ 0·4	+ 0·2	+ 0·1	+ 0	– 0·3	– 1·1	– 1·6	– 3·0
0·438– 0·5625	+ 0·5	+ 0·6	+ 1·0	+ 2·0	+ 1·0	+ 0·8	+ 0·6	+ 0·3	– 0	– 0·6	– 0·9	– 2·0
	+ 0	+ 0	+ 0	+ 0	+ 0·5	+ 0·3	+ 0·1	+ 0	– 0·3	– 1·2	– 1·9	– 4·0
0·563– 0·6875	+ 0·5	+ 0·7	+ 1·1	+ 2·0	+ 1·1	+ 0·8	+ 0·7	+ 0·3	– 0	– 0·7	– 1·0	– 2·0
	+ 0	+ 0	+ 0	+ 0	+ 0·6	+ 0·3	+ 0·2	+ 0	– 0·3	– 1·4	– 2·1	– 4·0
0·688– 0·8125	+ 0·5	+ 0·7	+ 1·2	+ 2·0	+ 1·3	+ 0·9	+ 0·7	+ 0·4	– 0	– 0·7	– 1·2	– 2·0
	+ 0	+ 0	+ 0	+ 0	+ 0·8	+ 0·4	+ 0·2	+ 0	– 0·4	– 1·4	– 2·4	– 4·0
0·813– 0·9375	+ 0·6	+ 0·8	+ 1·2	+ 2·0	+ 1·5	+ 1·0	+ 0·8	+ 0·4	– 0	– 0·8	– 1·3	– 2·0
	+ 0	+ 0	+ 0	+ 0	+ 0·9	+ 0·4	+ 0·2	+ 0	– 0·4	– 1·6	– 2·5	– 4·0
0·938– 1·0625	+ 0·6	+ 0·8	+ 1·3	+ 3·0	+ 1·6	+ 1·1	+ 0·9	+ 0·4	– 0	– 0·9	– 1·4	– 3·0
	+ 0	+ 0	+ 0	+ 0	+ 1·0	+ 0·5	+ 0·3	+ 0	– 0·4	– 1·7	– 2·7	– 6·0
1·063– 1·1875	+ 0·6	+ 0·8	+ 1·4	+ 3·0	+ 1·7	+ 1·2	+ 0·9	+ 0·4	– 0	– 1·0	– 1·5	– 3·0
	+ 0	+ 0	+ 0	+ 0	+ 1·1	+ 0·6	+ 0·3	+ 0	– 0·4	– 1·8	– 2·9	– 6·0
1·188– 1·375	+ 0·6	+ 0·9	+ 1·4	+ 3·0	+ 1·9	+ 1·2	+ 0·9	+ 0·4	– 0	– 1·0	– 1·6	– 3·0
	+ 0	+ 0	+ 0	+ 0	+ 1·3	+ 0·6	+ 0·3	+ 0	– 0·4	– 1·9	– 3·0	– 6·0
1·376– 1·625	+ 0·7	+ 0·9	+ 1·5	+ 3·0	+ 2·2	+ 1·5	+ 1·1	+ 0·5	– 0	– 1·2	– 1·8	– 3·0
	+ 0	+ 0	+ 0	+ 0	+ 1·5	+ 0·8	+ 0·4	+ 0	– 0·5	– 2·1	– 3·3	– 6·0
1·626– 1·875	+ 0·7	+ 1·0	+ 1·6	+ 3·0	+ 2·5	+ 1·6	+ 1·1	+ 0·5	– 0	– 1·3	– 2·0	– 4·0
	+ 0	+ 0	+ 0	+ 0	+ 1·8	+ 0·9	+ 0·4	+ 0	– 0·5	– 2·3	– 3·6	– 7·0
1·876– 2·125	+ 0·8	+ 1·0	+ 1·6	+ 3·0	+ 2·8	+ 1·8	+ 1·3	+ 0·5	– 0	– 1·4	– 2·2	– 4·0
	+ 0	+ 0	+ 0	+ 0	+ 2·0	+ 1·0	+ 0·5	+ 0	– 0·5	– 2·4	– 3·8	– 7·0
2·126– 2·375	+ 0·8	+ 1·0	+ 1·7	+ 3·0	+ 3·1	+ 1·9	+ 1·4	+ 0·5	– 0	– 1·5	– 2·4	– 4·0
	+ 0	+ 0	+ 0	+ 0	+ 2·3	+ 1·1	+ 0·6	+ 0	– 0·5	– 2·5	– 4·1	– 7·0
2·376– 2·75	+ 0·8	+ 1·1	+ 1·8	+ 3·0	+ 3·3	+ 2·1	+ 1·4	+ 0·5	– 0	– 1·7	– 2·6	– 5·0
	+ 0	+ 0	+ 0	+ 0	+ 2·5	+ 1·3	+ 0·6	+ 0	– 0·5	– 2·8	– 4·4	– 8·0
2·751– 3·25	+ 0·9	+ 1·2	+ 1·9	+ 4·0	+ 3·9	+ 2·4	+ 1·7	+ 0·6	– 0	– 1·9	– 2·9	– 5·0
	+ 0	+ 0	+ 0	+ 0	+ 3·0	+ 1·5	+ 0·8	+ 0	– 0·6	– 3·1	– 4·8	– 9·0
3·251– 3·75	+ 0·9	+ 1·2	+ 2·0	+ 4·0	+ 4·4	+ 2·7	+ 1·8	+ 0·6	– 0	– 2·1	– 3·2	– 6·0
	+ 0	+ 0	+ 0	+ 0	+ 3·5	+ 1·8	+ 0·9	+ 0	– 0·6	– 3·3	– 5·2	– 10·0
3·751– 4·25	+ 1·0	+ 1·3	+ 2·1	+ 4·0	+ 5·0	+ 3·0	+ 2·0	+ 0·6	– 0	– 2·3	– 3·5	– 6·0
	+ 0	+ 0	+ 0	+ 0	+ 4·0	+ 2·0	+ 1·0	+ 0	– 0·6	– 3·6	– 5·6	– 10·0
4·251– 4·75	+ 1·0	+ 1·3	+ 2·1	+ 4·0	+ 5·5	+ 3·3	+ 2·1	+ 0·7	– 0	– 2·5	– 3·8	– 7·0
	+ 0	+ 0	+ 0	+ 0	+ 4·5	+ 2·3	+ 1·1	+ 0	– 0·7	– 3·8	– 5·9	– 11·0
4·751– 5·5	+ 1·0	+ 1·4	+ 2·2	+ 4·0	+ 6·0	+ 3·5	+ 2·3	+ 0·7	– 0	– 2·6	– 4·1	– 7·0
	+ 0	+ 0	+ 0	+ 0	+ 5·0	+ 2·5	+ 1·3	+ 0	– 0·7	– 4·0	– 6·3	– 11·0
5·501– 6·5	+ 1·1	+ 1·5	+ 2·4	+ 5·0	+ 7·1	+ 4·1	+ 2·6	+ 0·7	– 0	– 3·0	– 4·6	– 8·0
	+ 0	+ 0	+ 0	+ 0	+ 6·0	+ 3·0	+ 1·5	+ 0	– 0·7	– 4·5	– 7·0	– 13·0
6·501– 7·5	+ 1·1	+ 1·5	+ 2·5	+ 5·0	+ 8·1	+ 4·6	+ 2·9	+ 0·8	– 0	– 3·3	– 5·1	– 9·0
	+ 0	+ 0	+ 0	+ 0	+ 7·0	+ 3·5	+ 1·8	+ 0	– 0·8	– 4·8	– 7·6	– 14·0
7·501– 8·5	+ 1·2	+ 1·6	+ 2·6	+ 5·0	+ 9·2	+ 5·2	+ 3·2	+ 0·8	– 0	– 3·6	– 5·6	– 10·0
	+ 0	+ 0	+ 0	+ 0	+ 8·0	+ 4·0	+ 2·0	+ 0	– 0·8	– 5·2	– 8·2	– 15·0
8·501–10·5	+ 1·2				+ 10·2	+ 5·7						
	+ 0				+ 9·0	+ 4·5						
10·501–11·5	+ 1·3				+ 11·3	+ 6·3						
	+ 0				+ 10·0	+ 5·0						
11·501–13	+ 1·4				+ 13·4	+ 7·4						
	+ 0				+ 12·0	+ 6·0						
13·001–15	+ 1·4				+ 15·4	+ 8·4						
	+ 0				+ 14·0	+ 7·0						
15·001–17	+ 1·5				+ 17·5	+ 9·5						
	+ 0				+ 16·0	+ 8·0						
17·001–19	+ 1·6				+ 19·6	+ 10·6						
	+ 0				+ 18·0	+ 9·0						
19·001–22	+ 1·6				+ 21·6	+ 11·6						
	+ 0				+ 20·0	+ 10·0						

TABLE 17

DERIVATION OF LIMITS

	Limits	Tolerance	Approx. Tolerance Ratio
Holes			
1.	$+ 0, + 0{\cdot}0006D^{\frac{1}{3}}$	$0{\cdot}0006D^{\frac{1}{3}}$	$1\frac{1}{2}$
2.	$+ 0, + 0{\cdot}0008D^{\frac{1}{3}}$	$0{\cdot}0008D^{\frac{1}{3}}$	2
3.	$+ 0, + 0{\cdot}0013D^{\frac{1}{3}}$	$0{\cdot}0013D^{\frac{1}{3}}$	3
4.	$+ 0, + 0{\cdot}0025D^{\frac{1}{3}}$	$0{\cdot}0025D^{\frac{1}{3}}$	6
Shafts			
Heavy Force	$+ (0{\cdot}001D + 0{\cdot}0006D^{\frac{1}{3}}), + 0{\cdot}001D$	$0{\cdot}0006D^{\frac{1}{3}}$	$1\frac{1}{2}$
Medium Force	$+ (0{\cdot}0005D + 0{\cdot}0006D^{\frac{1}{3}}), + 0{\cdot}0005D$	$0{\cdot}0006D^{\frac{1}{3}}$	$1\frac{1}{2}$
Tight	$+ (0{\cdot}00025D + 0{\cdot}0006D^{\frac{1}{3}}), + 0{\cdot}00025D$	$0{\cdot}0006D^{\frac{1}{3}}$	$1\frac{1}{2}$
Wringing	$+ 0{\cdot}0004D^{\frac{1}{3}}, + 0$	$0{\cdot}0004D^{\frac{1}{3}}$	1
Snug	$- 0, - 0{\cdot}0004D^{\frac{1}{3}}$	$0{\cdot}0004D^{\frac{1}{3}}$	1
Medium	$- 0{\cdot}0009D^{\frac{1}{3}}, - (0{\cdot}0009D^{\frac{1}{3}} + 0{\cdot}0008D^{\frac{1}{3}})$	$0{\cdot}0008D^{\frac{1}{3}}$	2
Free	$- 0{\cdot}0014D^{\frac{1}{3}}, - (0{\cdot}0014D^{\frac{1}{3}} + 0{\cdot}0013D^{\frac{1}{3}})$	$0{\cdot}0013D^{\frac{1}{3}}$	3
Loose	$- 0{\cdot}0025D^{\frac{1}{3}}, - (0{\cdot}0025D^{\frac{1}{3}} + 0{\cdot}0025D^{\frac{1}{3}})$	$0{\cdot}0025D^{\frac{1}{3}}$	6

The "Quality" or fundamental tolerance series to be compared with the I.S.A. system is thus—

$$1,\ 1\tfrac{1}{2},\ 2,\ 3,\ 6$$

Of particular interest when comparing one system with another are the formulae for the more important basic deviations (i.e. most important limits). Taking due note of the proper hole and shaft associations, these are given in Table 18.

TABLE 18

BASIC DEVIATION FORMULAE

Fit	Formulae	Remarks
Heavy Force	$0{\cdot}001D - 0{\cdot}0006D^{\frac{1}{3}}$	This is negative below 0·465 in. ($\frac{7}{16}$ in. nominal).
Medium Force	$0{\cdot}0005D - 0{\cdot}0006D^{\frac{1}{3}}$	This is negative below 1·32 in. ($1\frac{3}{16}$ in. nominal).
Tight	$0{\cdot}00025D - 0{\cdot}0006D^{\frac{1}{3}}$	This is negative below 3·74 in. ($3\frac{3}{4}$ in. nominal).
Snug	0	
Medium	$0{\cdot}0009D^{\frac{1}{3}}$	
Free	$0{\cdot}0014D^{\frac{1}{3}}$	
Loose	$0{\cdot}0025D^{\frac{1}{3}}$	

The change of sign in the three interference fits in the table in their smaller sizes will be noticed. Selective assembly will be needed to maintain the fits, and this is a bad fault.

5. Description of Fits

As indicated above, the fits are described and not referred to by symbols. The standard, however, lists additional explanatory advice on the intended uses of the fits.

Class	Proper Description	Recommended for
Interference*	Heavy Force and Shrink	Heavy shrink fits in steel holes, not cast iron.
	Medium Force	Heavy shrink fits on cast iron; permanently assembled parts; shrink fits on medium sections or long fits.
Transition	Tight	Press fit more or less permanent.
	Wringing	"Tunking" fit practically "metal to metal."
Clearance	Snug	"Used where no shake is permissible."
	Medium	Running fits under 600 r.p.m. with bearing pressures up to 600 lb/sq. in.
	Free	Running fits over 600 r.p.m. or 600 lb/sq. in.
	Loose	Considerable freedom where accuracy not essential.

6. Gauge Limits

No gauge limits have been published for the A.S.A. tolerance system.

THE I.S.A. TOLERANCE SYSTEM

The I.S.A. Tolerance System is the most modern of published tolerance systems and is certainly the most comprehensive and thorough. The system includes more than any other by formulating a system of tolerances, a system of limits and a system of gauges. It is the only system which lists several tolerance grades in each fit, and it is the only system which specifies gauging practice and gauge limits applicable to the tolerances involved.

Although the published system is very comprehensive, and indeed complicated, the practical application in a particular factory need be no more complicated than any other system, since a selection of fits to suit individual needs can be made.

The original metric system is published in I.S.A. Bulletin 25—the final issue being made in 1942, although earlier advance reports and publications had been in use on the Continent since about 1932.

The American Society of Mechanical Engineers published in

* Transition in small sizes

January, 1942, a slightly revised version of the English text (the original is in four languages) and converted the metric tolerance tables to inches for use with the inch system. This publication contained a number of errors and the conversion was not done in a correct manner. The tables which follow have been worked out in detail and published here for the first time, using the original I.S.A. formulae and rounded off or approximated in accordance with definite rules. The qualities of the various fits specified are identical with the metric original, although expressed in different units, but without involving the absurd nominal dimensions which a straight conversion would have entailed.

The original I.S.A. Bulletin 25 is a remarkably interesting document and well worth the study of the interested engineer.

1. Historical

The original proposal to commence work on a system of limits and fits was made at an I.S.A. meeting in New York in 1926. The Secretariat was entrusted to Germany, and a special sub-committee of experts from Czechoslovakia, France, Germany, Sweden, and Switzerland were responsible for the detail work. A preliminary report containing proposals for diameters 1–180 mm was published in 1932, and the final report in draft form, extended up to 500 mm, was issued in 1935.

2. Grades of Fit

The system consists fundamentally of 16 *qualities* of tolerance or, in other words, grade of manufacture; and of 20 grades of *fit* for both hole and shaft ranging from fits of extreme interference to fits of extreme clearance. All possible requirements of every range of engineering or scientific product are covered from very coarse manufacturing limits to those for an accuracy of manufacture approaching that achieved by gauge-block makers. It is intended that a particular industry or organization should extract and use those limits which suit its own product.

Although modern manufacturing practice is to use the "basic hole" or "unilateral hole" system with varying fits obtained by variation in shaft size, a full range of "bilateral" holes corresponding in fact to the shaft steps is also listed. These are frequently useful for special purposes (e.g. ball race housings).

The fits are designated by capital letters for holes (e.g. H) and small letters for shafts (e.g. f). A numerical suffix indicates the quality of tolerance in that particular hole or shaft (e.g. H8, f7, etc.). A fit is expressed by combining the symbols with the hole first, thus H7/g6 or H7–g6.

The various grades of fit which are recommended for normal work are shown diagrammatically in Fig. 72.

3. Association of Holes and Shafts

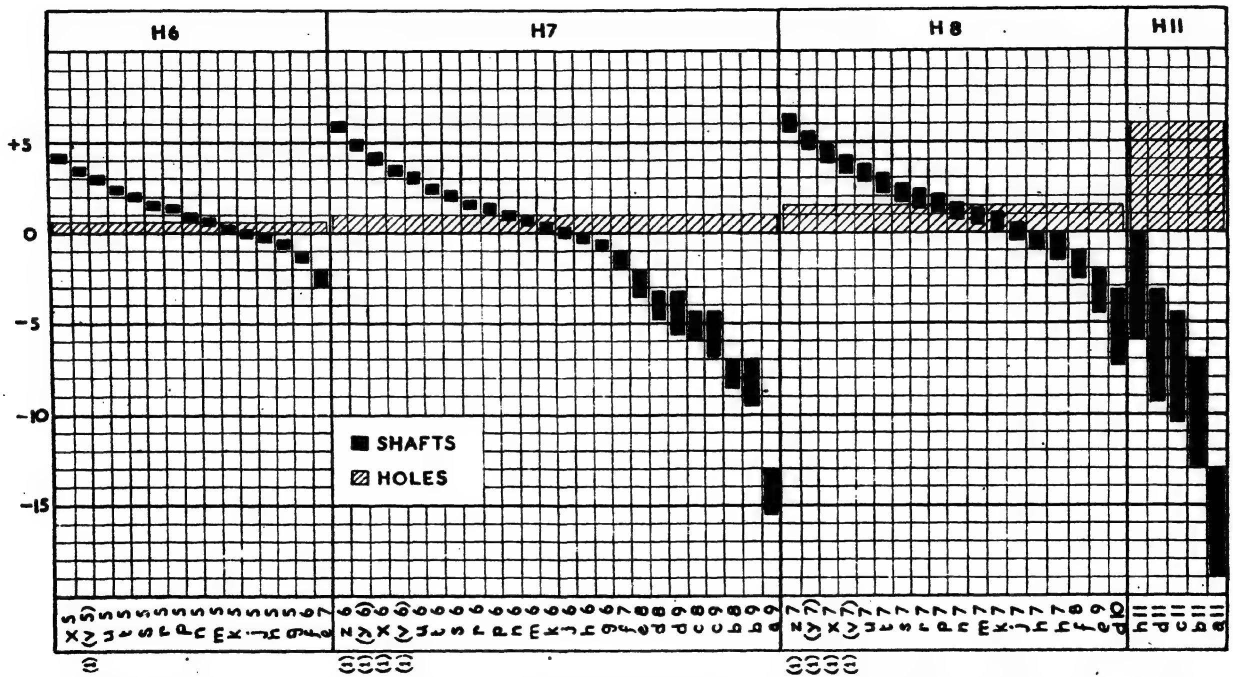
It is a principle of the system that any hole can be associated with any shaft in all the various fits and tolerances, but obviously many of the associations are most unlikely, and would never be made.

To obtain proper interference or clearance fits, certain associations are recommended as follows—

Press Fits			
Basic Hole	Associated Shafts	Basic Shaft	Associated Holes
H6	n5–x5	h5	N6–X6
H7	p6–z6	h6	R7–Z7
H8	s7–z7		

Clearance Fits							
Basic Hole	Associated Shafts						
H6	g5	f6	e7				
H7	g6	f7	e8	d8/d9	c8/c9	b8/b9	a9
H8		f8	e9	d10			
H11				d11	c11	b11	a11
Basic Shaft	Associated Holes						
h5	G6	F6	E7				
h6	G7	F7	E8	D8/D9	C8/C9*	B8/B9*	A9*
h8(h9)		F8	E9	D10			
h11				D11	C11	B11	A11

* Holes C8, C9, B8, B9, and A9 may be associated with shafts h7 and h8.



(1) The shafts *V*, *X*, *Y*, and *Z* are not to be considered as definite recommendations, but as recommended for trial. Shafts *V* and *Y* are to be avoided as much as possible

(Diagram to scale for Dia. 1·6 in.–2·0 in. unit 0.001 in.)

FIG. 72. I.S.A. TOLERANCE SYSTEM. RECOMMENDED SELECTION OF FITS FOR GENERAL ENGINEERING REQUIREMENTS

Fits on the basic hole system are equivalent to corresponding fits on the basic shaft system as indicated by the following—

Clearance Fits	Transition Fits	Interference Fits
H7/a9 = A9/h7	H6/j5 = J6/h5	H6/p5 = P6/h5
H11/a11 = A11/h11	H7/j6 = J7/h6	H7/p6 = P7/h6
H7/b8 = B8/h7	H8/j7 = J8/h7	H6/r5 = R6/h5
H7/b9 = B9/h7	H6/k5 = K6/h5	H7/r6 = R7/h6
H11/b11 = B11/h11	H7/h6 = K7/h6	H6/s5 = S6/h5
H7/c8 = C8/h7	H8/k7 = K8/h7	H7/s6 = S7/h6
H7/c9 = C9/h7	H6/m5 = M6/h5	H6/t5 = T6/h5
H11/c11 = C11/h11	H7/m6 = M7/h6	H7/t6 = T7/h6
H8/d10 = D10/h8	H8/m7 = M8/h7	H6/u5 = U6/h5
H11/d11 = D11/h11	H6/n5 = N6/h5	H7/u6 = U7/h6
H8/e9 = E9/h8	H7/n6 = N7/h6	H6/v5 = V6/h5
H8/f8 = F8/h8	H8/n7 = N8/h7	H7/v6 = V7/h6
H6/g5 = G6/h5		H6/x5 = X6/h5
H7/g6 = G7/h6		H7/x6 = X7/h6
H6/h5 = H6/h5		H7/y6 = Y7/h6
H7/h6 = H7/h6		H7/z6 = Z7/h6
H8/h7 = H8/h7		
H8/h8 = H8/h8		
H11/h11 = H11/h11		

4. Diameter Steps

The original metric version used diameter increments or steps based partly on Standard Numbers (10 and 20 series) and partly (in the sizes below 180 mm) on existing practice which could not be ignored.

These steps are as follows—

1, 3, 6, 10, 14, **18,** 24, **30,** 40, 50, 65, **80,** 100, **120,** 140, 160, **180,** 200, 225, **250,** 280, **315,** 355, **400,** 450, **500** mm.

In the translation, the steps are based on the same Standard Number series, but not having existing practice to take into account the steps are uncompromised throughout with the exception of the smallest sizes. The 10 series is used between 0·315 in. and 5 in., and the 20 series above 5 in. up to 20 in. Below 0·315 in. a single division chosen arbitrarily at 0·16 in. is taken between this size and 0·04 in., which is the smallest diameter tabled (1 mm in the original).

The diameter steps are as follows—

0·04, 0·16, 0·315, 0·4, **0·5,** 0·63, **0·80,** 1, **1·25,** 1·6, **2,** 2·5, **3·15,** 4, **5,** 5·6, **6·3,** 7·1, **8,** 9, **10,** 11·2, **12·5,** 14, **16,** 18, **20** inches.

The general or coarse steps are in heavy type. The intermediate or fine steps are used only in certain grades of hole or shaft with large clearance or interference.

5. Fundamental Tolerances

The 16 qualities of tolerance referred to in paragraph 2 are based on a *Fundamental Tolerance Unit* (i) and are multiples of it (IT1–IT16). Between IT6 and IT16 the tolerance units are arranged in a geometric series based on the 5-Series Standard Numbers.* The values of IT12 to IT16 are thus 10 times the values of IT6 to IT11. The value of IT5 is 70 per cent of IT6. The fundamental tolerance values of IT1 are arranged in arithmetic progression in a purely arbitrary fashion owing to the extreme difficulty in measuring extremely fine limits (the first value of IT1 is only 60 micro-inches). The values of IT2 to IT4 are arranged in an approximate geometric progression between IT1 and IT5.

The fundamental tolerance unit itself is derived from the formula—

$$i \text{ (microns)} = 0{\cdot}45\sqrt[3]{D} + 0{\cdot}001D \text{ (mm)}$$

or

$$i \text{ (0·001 in.)} = 0{\cdot}0520782\sqrt[3]{D} + 0{\cdot}001D \text{ (inches)}$$

where D is the geometric mean of the diameter steps involved. The term $0{\cdot}001D$ is introduced, to quote from I.S.A. Bulletin 25, "out of consideration for the uncertainties in measuring which increase in proportion to the increasing diameter (these being mainly due to differences in temperature and deformation of gauges and working pieces). In practice this is only noticeable in diameters over about 80 mm."

The fundamental tolerances (rounded values) are given in Tables 19 and 20.

6. Derivation of Limits

The fundamental tolerances determine the dimensional difference between two limits, but to determine the various fits one of the limits, i.e. the basic deviation, must be determined according to some law or formula, the other limit of course being obtained by adding or subtracting the fundamental tolerance. The full schedule of limits is set out in Tables 21 (holes) and 22 (shafts), Metric, and 23 (holes) and 24 (shafts), Inch system.

The fundamental limits of the holes (A–G) are identical with those of the shafts (a–g) but of opposite sign. The unilateral hole is H, A being + and Z −. Conversely, the shaft h has an upper limit of zero, a being − and z +.

The derivation of the various fundamental limits is as follows (D in all cases being the geometric mean of the diameter steps

* The number 64 was used because at the time the correct Standard Number 63 had not yet been accepted.

TABLE 19

FUNDAMENTAL TOLERANCES: METRIC UNITS

Unit = 1 micron = 0·001 mm

Diameter		For Gauges (IT1–IT7)							For Fits (IT5–IT11)				Large Tolerances (Not for Fits)				
Above	To	IT1	IT2	IT3	IT4	IT5	IT6	IT7	IT8	IT9	IT10	IT11	IT12	IT13	IT14	IT15	IT16
1	3	1·5	2	3	4	5	7	9	14	25	40	60	90	140	250	400	600
3	6	1·5	2	3	4	5	8	12	18	30	48	75	120	180	300	480	750
6	10	1·5	2	3	4	6	9	15	22	36	58	90	150	220	360	580	900
10	18	1·5	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100
18	30	1·5	2	4	6	9	13	21	33	52	84	130	210	330	520	840	1300
30	50	2	3	4	7	11	16	25	39	62	100	160	250	390	620	1000	1600
50	80	2	3	5	8	13	19	30	46	74	120	190	300	460	740	1200	1900
80	120	3	4	6	10	15	22	35	54	87	140	220	350	540	870	1400	2200
120	180	4	5	8	12	18	25	40	63	100	160	250	400	630	1000	1600	2500
180	250	5	7	10	14	20	29	46	72	115	185	290	460	720	1150	1850	2900
250	315	6	8	12	16	23	32	52	81	130	210	320	520	810	1300	2100	3200
315	400	7	9	13	18	25	36	57	89	140	230	360	570	890	1400	2300	3600
400	500	8	10	15	20	27	40	63	97	155	250	400	630	970	1550	2500	4000

TABLE 20

Fundamental Tolerances: Inch Units (Tolerances in 0·001 in.) (Rounded Off)

Nominal Diameters (inches)	Tolerance Unit = i	IT1	IT2	IT3	IT4	IT5	IT6	IT7	IT8	IT9	IT10	IT11	IT12	IT13	IT14	IT15	IT16
		For Gauges: IT1–IT7; For Fits: IT5–IT11											Large Tolerances (Not for Fits)				
		Grades															
Over 0·04 to 0·16	0·0225198	0·06	0·08	0·10	0·15	0·20*	0·25	0·35	0·55	0·90	1·4	2·3	3·6	5·5	9·0	14	23
Over 0·16 to 0·315	0·0318761	0·06	0·08	0·10	0·15	0·20	0·30	0·50	0·80	1·3	2·0	3·2	5·0	8·0	13	20	32
Over 0·315 to 0·5	0·0386679	0·06	0·08	0·10	0·15	0·25	0·40	0·60	0·95	1·5	2·5	3·9	6·0	9·5	15	25	39
Over 0·5 to 0·8	0·0453351	0·06	0·08	0·10	0·20	0·30	0·45	0·75	1·1	1·8	2·9	4·5	7·5	11	18	29	45
Over 0·8 to 1·25	0·0530783	0·06	0·08	0·15	0·25	0·35	0·55	0·85	1·3	2·1	3·4	5·5	8·5	13	21	34	53
Over 1·25 to 2·0	0·0622520	0·08	0·12	0·15	0·30	0·45	0·60	1·0	1·6	2·5	4·0	6·0	10	16	25	40	62
Over 2·0 to 3·15	0·0732850	0·08	0·12	0·20	0·35	0·50	0·75	1·2	1·8	2·9	4·7	7·5	12	18	29	47	73
Over 3·15 to 5·0	0·0864211	0·12	0·16	0·25	0·40	0·60	0·85	1·4	2·2	3·5	5·5	8·5	14	22	35	55	86
Over 5·0 to 6·3	0·0981622	0·12	0·20	0·30	0·45	0·70	1·0	1·6	2·5	3·9	6·5	10	16	25	39	62	98
Over 6·3 to 8·0	0·107190	0·16	0·24	0·35	0·50	0·75	1·1	1·7	2·7	4·3	7·0	11	17	27	43	69	107
Over 8·0 to 10·0	0·117047	0·20	0·28	0·40	0·55	0·80	1·2	1·9	2·9	4·7	7·5	12	19	29	47	75	117
Over 10·0 to 12·5	0·127639	0·24	0·32	0·45	0·65	0·90	1·3	2·0	3·2	5·0	8·0	13	20	32	51	82	128
Over 12·5 to 16·0	0·140082	0·28	0·36	0·55	0·75	1·0	1·4	2·2	3·5	5·5	9·0	14	32	35	56	90	140
Over 16·0 to 20·0	0·154090	0·32	0·40	0·65	0·85	1·1	1·5	2·5	3·9	6·0	10	15	25	39	62	99	154
Rounded off to:				←0·05	→	←	0·1		→	←0·5→	←			0·1			→

* Rounded to 0·20 instead of 0·15 for convenience.

involved). The mm dimensions in brackets refer to the metric I.S.A. figures converted approximately in some cases.

In the case of classes a, A to h, H, the formulae give the upper limit of shafts and the lower limit of holes. In the case of classes k, K to z, Z, the formulae give the lower limit of shafts.

(i) a, A $= 10{\cdot}5$ (265 microns) $+ 1{\cdot}3D$ (when $D \leq 5{\cdot}0$) (120 mm)
$= 3{\cdot}5D$ (when $D > 5{\cdot}0$) (120 mm)

(ii) b, B $= 5{\cdot}5$ (140 microns) $+ 0{\cdot}85D$ (when $D \leq 5{\cdot}0$) (120 mm)
$= 1{\cdot}8D$ (when $D > 5{\cdot}0$) (120 mm)

(iii) c, C $= 3{\cdot}909636D^{0{\cdot}2}$ ($52D^{0{\cdot}2}$ microns) (when $D \leq 2{\cdot}0$) (40 mm)
$= 3{\cdot}75$ (95 microns) $+ 0{\cdot}8D$ (when $D > 2{\cdot}0$) (40 mm)

(iv) d, D $= 2{\cdot}616454D^{0{\cdot}44}$ ($16D^{0{\cdot}44}$ in. mm)

(v) e, E $= 1{\cdot}631326D^{0{\cdot}41}$ ($11D^{0{\cdot}41}$ in. mm)

(vi) f, F $= 0{\cdot}815663D^{0{\cdot}41}$ ($5{\cdot}5D^{0{\cdot}41}$ in. mm)

(vii) g, G $= 0{\cdot}295631D^{0{\cdot}34}$ ($2{\cdot}5D^{0{\cdot}34}$ in. mm)

(viii) h, H $= 0$

(ix) j, J. For qualities 5, 6, and 7, both shaft limits are determined in a purely arbitrary fashion based on experience.
The lower limits of the holes J6, J7, and J8 are the same as the upper limits of shafts j5, j6, and j7, respectively, but of opposite sign.
For j8 and J9 and after, the fundamental tolerance is divided equally + and −. To avoid awkward division when the fundamental tolerance is an odd number, the + of the shaft and the − of the hole is given the greater integral number (e.g. $15 = 8 + 7$).

(x) k. For qualities 5, 6, and 7, the lower limit of the shaft is determined according to the formula $k = 0{\cdot}069438 \sqrt[3]{D}$ ($0{\cdot}6 \sqrt[3]{D}$ in. mm). For qualities 8, 9, 10, and 11, the lower limits of the shafts is zero, but these are not intended for fits.

K. The lower limits of the holes K6, K7, K8 (the only ones listed) are the same as the upper limits of shafts k5, k6, k7, respectively, but of opposite sign. Some of the values have been adjusted slightly for the sake of the progression, the addition of several rounded-off figures causing anomalies. (See also Note (*c*) in para. 7 below.)

(xi) m $= 0{\cdot}324043 \sqrt[3]{D}$ ($2{\cdot}8 \sqrt[3]{D}$ in. mm).
In the case of m6 the upper limit is made to correspond with the hole H7, making the lower limit of m6 depart from the theoretical as well.

M. The lower limits of the holes M6, M7, and M8 are the same as the upper limits of shafts m5, m6, and m7, respectively, but of opposite sign, and adjusted as for K.

(xii) n $= 0{\cdot}591262D^{0{\cdot}34}$ ($= 5D^{0{\cdot}34}$ in. mm), with the overriding consideration that it must be greater than H6 except in the smallest sizes.

N. The lower limits of the holes N6, N7, and N8 are the same as the upper limits of shafts n5, n6, and n7, respectively, but of opposite sign. The limits of N9, N10, and N11 correspond exactly with shafts h9, h10, and h11, and are adjusted as for K.

(xiii) p. The lower limit was determined more or less arbitrarily as the same size or a few microns larger than the upper limit of H7, since this shaft was intended as the first true interference fit. It exists in qualities 5, 6, and 7 only.

P. The lower limits of the holes P6 and P7 are the same as the upper limits of the shafts p5 and p6, respectively, but of opposite sign.

(xiv) r. The lower limit is the geometric mean of the corresponding p and s limits.

R. The limits for R6 and R7 are derived from r5 and r6 as for P.

(xv) s. The limits of the shaft always produce an interference when used with the hole H8. Up to and including diameters of 2·0 in. (50 mm), the fundamental limits have been arranged arbitrarily in this way. For diameters above 2·0 in. (50 mm)

$$s = 0{\cdot}4D + \mathrm{IT7}$$

S. Derived as for R.

(xvi) t. As for s up to diameter 2·0 in. (50 mm).
For diameters above 2·0 in. (50 mm).

$$\mathrm{t} = 0{\cdot}63D + \mathrm{IT7}$$

T. Derived as for R.

(xvii) u. As for s up to diameter 0·8 in. (18 mm).
For diameters above 0·8 in. (18 mm).

$$\mathrm{u} = 1{\cdot}0D + \mathrm{IT7}$$

U. Derived as for R.

(xviii) v. As for s up to diameter 0·8 in. (18 mm).
For diameters above 0·8 in. (18 mm).

$$\mathrm{v} = 1{\cdot}25D + \mathrm{IT7}$$

V. Derived as for R.

(xix) x. As for s up to diameter 0·8 in. (18 mm).
For diameters above 0·8 in. (18 mm).

$$\mathrm{x} = 1{\cdot}6D + \mathrm{IT7}$$

X. Derived as for R.

(xx) y. As for s up to diameter 0·8 in. (18 mm).
For diameters above 0·8 in. (18 mm).

$$\mathrm{y} = 2{\cdot}0D + \mathrm{IT7}$$

Y. Derived as for R.

(xxi) z. As for s up to diameter 0·8 in. (18 mm).
For diameters above 0·8 in. (18 mm).

$$\mathrm{z} = 2{\cdot}5D + \mathrm{IT7}$$

Z. Derived as for R.

TABLE 21. I.S.A. LIMITS—

Quality 6 — HOLES — Limits in Microns

Nominal Size, mm	F6	G6	H6	J6	K6	M6
1·01– 3	+ 14	+ 10	+ 7	+ 3	—	− 0
	+ 7	+ 3	+ 0	− 4	—	− 7
3·01– 6	+ 18	+ 12	+ 8	+ 4	—	− 1
	+ 10	+ 4	+ 0	− 4	—	− 9
6·01– 10	+ 22	+ 14	+ 9	+ 5	+ 2	− 3
	+ 13	+ 5	+ 0	− 4	− 7	− 12
10·01– 14	+ 27	+ 17	+ 11	+ 6	+ 2	− 4
	+ 16	+ 6	+ 0	− 5	− 9	− 15
14·01– 18	+ 27	+ 17	+ 11	+ 6	+ 2	− 4
	+ 16	+ 6	+ 0	− 5	− 9	− 15
18·01– 24	+ 33	+ 20	+ 13	+ 8	+ 2	− 4
	+ 20	+ 7	+ 0	− 5	− 11	− 17
24·01– 30	+ 33	+ 20	+ 13	+ 8	+ 2	− 4
	+ 20	+ 7	+ 0	− 5	− 11	− 17
30·01– 40	+ 41	+ 25	+ 16	+ 10	+ 3	− 4
	+ 25	+ 9	+ 0	− 6	− 13	− 20
40·01– 50	+ 41	+ 25	+ 16	+ 10	+ 3	− 4
	+ 25	+ 9	+ 0	− 6	− 13	− 20
50·01– 65	+ 49	+ 29	+ 19	+ 13	+ 4	− 5
	+ 30	+ 10	+ 0	− 6	− 15	− 24
65·01– 80	+ 49	+ 29	+ 19	+ 13	+ 4	− 5
	+ 30	+ 10	+ 0	− 6	− 15	− 24
80·01–100	+ 58	+ 34	+ 22	+ 16	+ 4	− 6
	+ 36	+ 12	+ 0	− 6	− 18	− 28
100·1 –120	+ 58	+ 34	+ 22	+ 16	+ 4	− 6
	+ 36	+ 12	+ 0	− 6	− 18	− 28
120·1 –140	+ 68	+ 39	+ 25	+ 18	+ 4	− 8
	+ 43	+ 14	+ 0	− 7	− 21	− 33
140·1 –160	+ 68	+ 39	+ 25	+ 18	+ 4	− 8
	+ 43	+ 14	+ 0	− 7	− 21	− 33
160·1 –180	+ 68	+ 39	+ 25	+ 18	+ 4	− 8
	+ 43	+ 14	+ 0	− 7	− 21	− 33
180·1 –200	+ 79	+ 44	+ 29	+ 22	+ 5	− 8
	+ 50	+ 15	+ 0	− 7	− 24	− 37
200·1 –225	+ 79	+ 44	+ 29	+ 22	+ 5	− 8
	+ 50	+ 15	+ 0	− 7	− 24	− 37
225·1 –250	+ 79	+ 44	+ 29	+ 22	+ 5	− 8
	+ 50	+ 15	+ 0	− 7	− 24	− 37
250·1 –280	+ 88	+ 49	+ 32	+ 25	+ 5	− 9
	+ 56	+ 17	+ 0	− 7	− 27	− 41
280·1 –315	+ 88	+ 49	+ 32	+ 25	+ 5	− 9
	+ 56	+ 17	+ 0	− 7	− 27	− 41
315·1 –355	+ 98	+ 54	+ 36	+ 29	+ 7	− 10
	+ 62	+ 18	+ 0	− 7	− 29	− 46
355·1 –400	+ 98	+ 54	+ 36	+ 29	+ 7	− 10
	+ 62	+ 18	+ 0	− 7	− 29	− 46
400·1 –450	+ 108	+ 60	+ 40	+ 33	+ 8	− 10
	+ 68	+ 20	+ 0	− 7	− 32	− 50
450·1 –500	+ 108	+ 60	+ 40	+ 33	+ 8	− 10
	+ 68	+ 20	+ 0	− 7	− 32	− 50

HOLES: METRIC SYSTEM
(μ = 0·001 mm)

N6	P6	R6	S6	T6	U6	(V6)	X6
− 4	− 7	− 10	− 13	—	− 16	—	− 20
− 11	− 14	− 17	− 20	—	− 23	—	− 27
− 5	− 9	− 12	− 16	—	− 20	—	− 25
− 13	− 17	− 20	− 24	—	− 28	—	− 33
− 7	− 12	− 16	− 20	—	− 25	—	− 31
− 16	− 21	− 25	− 29	—	− 34	—	− 40
− 9	− 15	− 20	− 25	—	− 30	—	− 37
− 20	− 26	− 31	− 36	—	− 41	—	− 48
− 9	− 15	− 20	− 25	—	− 30	− 36	− 42
− 20	− 26	− 31	− 36	—	− 41	− 47	− 53
− 11	− 18	− 24	− 31	—	− 37	− 43	− 50
− 24	− 31	− 37	− 44	—	− 50	− 56	− 63
− 11	− 18	− 24	− 31	− 37	− 44	− 51	− 60
− 24	− 31	− 37	− 44	− 50	− 57	− 64	− 73
− 12	− 21	− 29	− 38	− 43	− 55	− 63	− 75
− 28	− 37	− 45	− 54	− 59	− 71	− 79	− 91
− 12	− 21	− 29	− 38	− 49	− 65	− 76	− 92
− 28	− 37	− 45	− 54	− 65	− 81	− 92	− 108
− 14	− 26	− 35	− 47	− 60	− 81	− 96	− 116
− 33	− 45	− 54	− 66	− 79	− 100	− 115	− 135
− 14	− 26	− 37	− 53	− 69	− 96	− 114	− 140
− 33	− 45	− 56	− 72	− 88	− 115	− 133	− 159
− 16	− 30	− 44	− 64	− 84	− 117	− 139	− 171
− 38	− 52	− 66	− 86	− 106	− 139	− 161	− 193
− 16	− 30	− 47	− 72	− 97	− 137	− 165	− 203
− 38	− 52	− 69	− 94	− 119	− 159	− 187	− 225
− 20	− 36	− 56	− 85	− 115	− 163	− 195	− 241
− 45	− 61	− 81	− 110	− 140	− 188	− 220	− 266
− 20	− 36	− 58	− 93	− 127	− 183	− 221	− 273
− 45	− 61	− 83	− 118	− 152	− 208	− 246	− 298
− 20	− 36	− 61	− 101	− 139	− 203	− 245	− 303
− 45	− 61	− 86	− 126	− 164	− 228	− 270	− 328
− 22	− 41	− 68	− 113	− 157	− 227	− 275	− 341
− 51	− 70	− 97	− 142	− 186	− 256	− 304	− 370
− 22	− 41	− 71	− 121	− 171	− 249	− 301	− 376
− 51	− 70	− 100	− 150	− 200	− 278	− 330	− 405
− 22	− 41	− 75	− 131	− 187	− 275	− 331	− 416
− 51	− 70	− 104	− 160	− 216	− 304	− 360	− 445
− 25	− 47	− 85	− 149	− 209	− 306	− 376	− 466
− 57	− 79	− 117	− 181	− 241	− 338	− 408	− 498
− 25	− 47	− 89	− 161	− 231	− 341	− 416	− 516
− 57	− 79	− 121	− 193	− 263	− 373	− 448	− 548
− 26	− 51	− 97	− 179	− 257	− 379	− 464	− 579
− 62	− 87	− 133	− 215	− 293	− 415	− 500	− 615
− 26	− 51	− 103	− 197	− 283	− 424	− 519	− 649
− 62	− 87	− 139	− 233	− 319	− 460	− 555	− 685
− 27	− 55	− 113	− 219	− 317	− 477	− 582	− 727
− 67	− 95	− 153	− 259	− 357	− 517	− 622	− 767
− 27	− 55	− 119	− 239	− 347	− 527	− 647	− 807
− 67	− 95	− 159	− 279	− 387	− 567	− 687	− 847

TABLE 21 (*contd.*). I.S.A. LIMITS–

Quality 7 HOLES Limits in Micron

Nominal Size, mm	E7	F7	G7	H7	J7	K7	M7	N7
1·01– 3	+ 23	+ 16	+ 12	+ 9	+ 3	—	− 0	− 4
	+ 14	+ 7	+ 3	+ 0	− 6	—	− 9	− 13
3·01– 6	+ 32	+ 22	+ 16	+ 12	+ 5	—	− 0	− 4
	+ 20	+ 10	+ 4	+ 0	− 7	—	− 12	− 16
6·01– 10	+ 40	+ 28	+ 20	+ 15	+ 8	+ 5	− 0	− 4
	+ 25	+ 13	+ 5	+ 0	− 7	− 10	− 15	− 19
10·01– 14	+ 50	+ 34	+ 24	+ 18	+ 10	+ 6	− 0	− 5
	+ 32	+ 16	+ 6	+ 0	− 8	− 12	− 18	− 23
14·01– 18	+ 50	+ 34	+ 24	+ 18	+ 10	+ 6	− 0	− 5
	+ 32	+ 16	+ 6	+ 0	− 8	− 12	− 18	− 23
18·01– 24	+ 61	+ 41	+ 28	+ 21	+ 12	+ 6	− 0	− 7
	+ 40	+ 20	+ 7	+ 0	− 9	− 15	− 21	− 28
24·01– 30	+ 61	+ 41	+ 28	+ 21	+ 12	+ 6	− 0	− 7
	+ 40	+ 20	+ 7	+ 0	− 9	− 15	− 21	− 28
30·01– 40	+ 75	+ 50	+ 34	+ 25	+ 14	+ 7	− 0	− 8
	+ 50	+ 25	+ 9	+ 0	− 11	− 18	− 25	− 33
40·01– 50	+ 75	+ 50	+ 34	+ 25	+ 14	+ 7	− 0	− 8
	+ 50	+ 25	+ 9	+ 0	− 11	− 18	− 25	− 33
50·01– 65	+ 90	+ 60	+ 40	+ 30	+ 18	+ 9	− 0	− 9
	+ 60	+ 30	+ 10	+ 0	− 12	− 21	− 30	− 39
65·01– 80	+ 90	+ 60	+ 40	+ 30	+ 18	+ 9	− 0	− 9
	+ 60	+ 30	+ 10	+ 0	− 12	− 21	− 30	− 39
80·01–100	+ 107	+ 71	+ 47	+ 35	+ 22	+ 10	− 0	− 10
	+ 72	+ 36	+ 12	+ 0	− 13	− 25	− 35	− 45
100·1 –120	+ 107	+ 71	+ 47	+ 35	+ 22	+ 10	− 0	− 10
	+ 72	+ 36	+ 12	+ 0	− 13	− 25	− 35	− 45
120·1 –140	+ 125	+ 83	+ 54	+ 40	+ 26	+ 12	− 0	− 12
	+ 85	+ 43	+ 14	+ 0	− 14	− 28	− 40	− 52
140·1 –160	+ 125	+ 83	+ 54	+ 40	+ 26	+ 12	− 0	− 12
	+ 85	+ 43	+ 14	+ 0	− 14	− 28	− 40	− 52
160·1 –180	+ 125	+ 83	+ 54	+ 40	+ 26	+ 12	− 0	− 12
	+ 85	+ 43	+ 14	+ 0	− 14	− 28	− 40	− 52
180·1 –200	+ 146	+ 96	+ 61	+ 46	+ 30	+ 13	− 0	− 14
	+ 100	+ 50	+ 15	+ 0	− 16	− 33	− 46	− 60
200·1 –225	+ 146	+ 96	+ 61	+ 46	+ 30	+ 13	− 0	− 14
	+ 100	+ 50	+ 15	+ 0	− 16	− 33	− 46	− 60
225·1 –250	+ 146	+ 96	+ 61	+ 46	+ 30	+ 13	− 0	− 14
	+ 100	+ 50	+ 15	+ 0	− 16	− 33	− 46	− 60
250·1 –280	+ 162	+ 108	+ 69	+ 52	+ 36	+ 16	− 0	− 14
	+ 110	+ 56	+ 17	+ 0	− 16	− 36	− 52	− 66
280·1 –315	+ 162	+ 108	+ 69	+ 52	+ 36	+ 16	− 0	− 14
	+ 110	+ 56	+ 17	+ 0	− 16	− 36	− 52	− 66
315·1 –355	+ 182	+ 119	+ 75	+ 57	+ 39	+ 17	− 0	− 16
	+ 125	+ 62	+ 18	+ 0	− 18	− 40	− 57	− 73
355·1 –400	+ 182	+ 119	+ 75	+ 57	+ 39	+ 17	− 0	− 16
	+ 125	+ 62	+ 18	+ 0	− 18	− 40	− 57	− 73
400·1 –450	+ 198	+ 131	+ 83	+ 63	+ 43	+ 18	− 0	− 17
	+ 135	+ 68	+ 20	+ 0	− 20	− 45	− 63	− 80
450·1 –500	+ 198	+ 131	+ 83	+ 63	+ 43	+ 18	− 0	− 17
	+ 135	+ 68	+ 20	+ 0	− 20	− 45	− 63	− 80

HOLES: METRIC SYSTEM

(μ = 0·001 mm)

P7	R7	S7	T7	U7	(V7)	X7	(Y7)	Z7
− 7	− 10	− 13	—	− 16	—	− 20	—	− 26
− 16	− 19	− 22	—	− 25	—	− 29	—	− 35
− 8	− 11	− 15	—	− 19	—	− 24	—	− 31
− 20	− 23	− 27	—	− 31	—	− 36	—	− 43
− 9	− 13	− 17	—	− 22	—	− 28	—	− 36
− 24	− 28	− 32	—	− 37	—	− 43	—	− 51
− 11	− 16	− 21	—	− 26	—	− 33	—	− 43
− 29	− 34	− 39	—	− 44	—	− 51	—	− 61
− 11	− 16	− 21	—	− 26	− 32	− 38	—	− 53
− 29	− 34	− 39	—	− 44	− 50	− 56	—	− 71
− 14	− 20	− 27	—	− 33	− 39	− 46	− 55	− 65
− 35	− 41	− 48	—	− 54	− 60	− 67	− 76	− 86
− 14	− 20	− 27	− 33	− 40	− 47	− 56	− 67	− 80
− 35	− 41	− 48	− 54	− 61	− 68	− 77	− 88	− 101
− 17	− 25	− 34	− 39	− 51	− 59	− 71	− 85	− 103
− 42	− 50	− 59	− 64	− 76	− 84	− 96	− 110	− 128
− 17	− 25	− 34	− 45	− 61	− 72	− 88	− 105	− 127
− 42	− 50	− 59	− 70	− 86	− 97	− 113	− 130	− 152
− 21	− 30	− 42	− 55	− 76	− 91	− 111	− 133	− 161
− 51	− 60	− 72	− 85	− 106	− 121	− 141	− 163	− 191
− 21	− 32	− 48	− 64	− 91	− 109	− 135	− 163	− 199
− 51	− 62	− 78	− 94	− 121	− 139	− 165	− 193	− 229
− 24	− 38	− 58	− 78	− 111	− 133	− 165	− 201	− 245
− 59	− 73	− 93	− 113	− 146	− 168	− 200	− 236	− 280
− 24	− 41	− 66	− 91	− 131	− 159	− 197	− 241	− 297
− 59	− 76	− 101	− 126	− 166	− 194	− 232	− 276	− 332
− 28	− 48	− 77	− 107	− 155	− 187	− 233	− 285	− 350
− 68	− 88	− 117	− 147	− 195	− 227	− 273	− 325	− 390
− 28	− 50	− 85	− 119	− 175	− 213	− 265	− 325	− 400
− 68	− 90	− 125	− 159	− 215	− 253	− 305	− 365	− 440
− 28	− 53	− 93	− 131	− 195	− 237	− 295	− 365	− 450
− 68	− 93	− 133	− 171	− 235	− 277	− 335	− 405	− 490
− 33	− 60	− 105	− 149	− 219	− 267	− 333	− 408	− 503
− 79	− 106	− 151	− 195	− 265	− 313	− 379	− 454	− 549
− 33	− 63	− 113	− 163	− 241	− 293	− 368	− 453	− 558
− 79	− 109	− 159	− 209	− 287	− 339	− 414	− 499	− 604
− 33	− 67	− 123	− 179	− 267	− 323	− 408	− 503	− 623
− 79	− 113	− 169	− 225	− 313	− 369	− 454	− 549	− 669
− 36	− 74	− 138	− 198	− 295	− 365	− 455	− 560	− 690
− 88	− 126	− 190	− 250	− 347	− 417	− 507	− 612	− 742
− 36	− 78	− 150	− 220	− 330	− 405	− 505	− 630	− 770
− 88	− 130	− 202	− 272	− 382	− 457	− 557	− 682	− 822
− 41	− 87	− 169	− 247	− 369	− 454	− 569	− 709	− 879
− 98	− 144	− 226	− 304	− 426	− 511	− 626	− 766	− 936
− 41	− 93	− 187	− 273	− 414	− 509	− 639	− 799	− 979
− 98	− 150	− 244	− 330	− 471	− 566	− 696	− 856	− 1036
− 45	− 103	− 209	− 307	− 467	− 572	− 717	− 897	− 1077
− 108	− 166	− 272	− 370	− 530	− 635	− 780	− 960	− 1140
− 45	− 109	− 229	− 337	− 517	− 637	− 797	− 977	− 1227
− 108	− 172	− 292	− 400	− 580	− 700	− 860	− 1040	− 1290

TABLE 21 (*contd.*)

I.S.A. Limits—Holes: Metric System

Quality 8 HOLES Limits in Microns (μ = 0·001 mm)

Nominal Size, mm	B8	C8	D8	E8	F8	H8	J8	K8	M8	N8
1·01– 3	+ 154	+ 74	+ 34	+ 28	+ 21	+ 14	+ 7	—	—	− 1
	+ 140	+ 60	+ 20	+ 14	+ 7	+ 0	− 7	—	—	− 15
3·01– 6	+ 158	+ 88	+ 48	+ 38	+ 28	+ 18	+ 9	—	—	− 2
	+ 140	+ 70	+ 30	+ 20	+ 10	+ 0	− 9	—	—	− 20
6·01– 10	+ 172	+ 102	+ 62	+ 47	+ 35	+ 22	+ 12	+ 6	+ 1	− 3
	+ 150	+ 80	+ 40	+ 25	+ 13	+ 0	− 10	− 16	− 21	− 25
10·01– 14	+ 177	+ 122	+ 77	+ 59	+ 43	+ 27	+ 15	+ 8	+ 2	− 3
	+ 150	+ 95	+ 50	+ 32	+ 16	+ 0	− 12	− 19	− 25	− 30
14·01– 18	+ 177	+ 122	+ 77	+ 59	+ 43	+ 27	+ 15	+ 8	+ 2	− 3
	+ 150	+ 95	+ 50	+ 32	+ 16	+ 0	− 12	− 19	− 25	− 30
18·01– 24	+ 193	+ 143	+ 98	+ 73	+ 53	+ 33	+ 20	+ 10	+ 4	− 3
	+ 160	+ 110	+ 65	+ 40	+ 20	+ 0	− 13	− 23	− 29	− 36
24·01– 30	+ 193	+ 143	+ 98	+ 73	+ 53	+ 33	+ 20	+ 10	+ 4	− 3
	+ 160	+ 110	+ 65	+ 40	+ 20	+ 0	− 13	− 23	− 29	− 36
30·01– 40	+ 209	+ 159	+ 119	+ 89	+ 64	+ 39	+ 24	+ 12	+ 5	− 3
	+ 170	+ 120	+ 80	+ 50	+ 25	+ 0	− 15	− 27	− 34	− 42
40·01– 50	+ 219	+ 169	+ 119	+ 89	+ 64	+ 39	+ 24	+ 12	+ 5	− 3
	+ 180	+ 130	+ 80	+ 50	+ 25	+ 0	− 15	− 27	− 34	− 42
50·01– 65	+ 236	+ 186	+ 146	+ 106	+ 76	+ 46	+ 28	+ 14	+ 5	− 4
	+ 190	+ 140	+ 100	+ 60	+ 30	+ 0	− 18	− 32	− 41	− 50
65·01– 80	+ 246	+ 196	+ 146	+ 106	+ 76	+ 46	+ 28	+ 14	+ 5	− 4
	+ 200	+ 150	+ 100	+ 60	+ 30	+ 0	− 18	− 32	− 41	− 50
80·01–100	+ 274	+ 224	+ 174	+ 126	+ 90	+ 54	+ 34	+ 16	+ 6	− 4
	+ 220	+ 170	+ 120	+ 72	+ 36	+ 0	− 20	− 38	− 48	− 58
100·1 –120	+ 294	+ 234	+ 174	+ 126	+ 90	+ 54	+ 34	+ 16	+ 6	− 4
	+ 240	+ 180	+ 120	+ 72	+ 36	+ 0	− 20	− 38	− 48	− 58
120·1 –140	+ 323	+ 263	+ 208	+ 148	+ 106	+ 63	+ 41	+ 20	+ 8	− 4
	+ 260	+ 200	+ 145	+ 85	+ 43	+ 0	− 22	− 43	− 55	− 67
140·1 –160	+ 343	+ 273	+ 208	+ 148	+ 106	+ 63	+ 41	+ 20	+ 8	− 4
	+ 280	+ 210	+ 145	+ 85	+ 43	+ 0	− 22	− 43	− 55	− 67
160·1 –180	+ 373	+ 293	+ 208	+ 148	+ 106	+ 63	+ 41	+ 20	+ 8	− 4
	+ 310	+ 230	+ 145	+ 85	+ 43	+ 0	− 22	− 43	− 55	− 67
180·1 –200	+ 412	+ 312	+ 242	+ 172	+ 122	+ 72	+ 47	+ 22	+ 9	− 5
	+ 340	+ 240	+ 170	+ 100	+ 50	+ 0	− 25	− 50	− 63	− 77
200·1 –225	+ 452	+ 332	+ 242	+ 172	+ 122	+ 72	+ 47	+ 22	+ 9	− 5
	+ 380	+ 260	+ 170	+ 100	+ 50	+ 0	− 25	− 50	− 63	− 77
225·1 –250	+ 492	+ 352	+ 242	+ 172	+ 122	+ 72	+ 47	+ 22	+ 9	− 5
	+ 420	+ 280	+ 170	+ 100	+ 50	+ 0	− 25	− 50	− 63	− 77
250·1 –280	+ 561	+ 381	+ 271	+ 191	+ 137	+ 81	+ 55	+ 25	+ 9	− 5
	+ 480	+ 300	+ 190	+ 110	+ 56	+ 0	− 26	− 56	− 72	− 86
280·1 –315	+ 621	+ 411	+ 271	+ 191	+ 137	+ 81	+ 55	+ 25	+ 9	− 5
	+ 540	+ 330	+ 190	+ 110	+ 56	+ 0	− 26	− 56	− 72	− 86
315·1 –355	+ 689	+ 449	+ 299	+ 214	+ 151	+ 89	+ 60	+ 28	+ 11	− 5
	+ 600	+ 360	+ 210	+ 125	+ 62	+ 0	− 29	− 61	− 78	− 94
355·1 –400	+ 769	+ 489	+ 299	+ 214	+ 151	+ 89	+ 60	+ 28	+ 11	− 5
	+ 680	+ 400	+ 210	+ 125	+ 62	+ 0	− 29	− 61	− 78	− 94
400·1 –450	+ 857	+ 537	+ 327	+ 232	+ 165	+ 97	+ 66	+ 29	+ 11	− 6
	+ 760	+ 440	+ 230	+ 135	+ 68	+ 0	− 31	− 68	− 86	− 103
450·1 –500	+ 937	+ 577	+ 327	+ 232	+ 165	+ 97	+ 66	+ 29	+ 11	− 6
	+ 840	+ 480	+ 230	+ 135	+ 68	+ 0	− 31	− 68	− 86	− 103

TABLE 21 (*contd.*)

I.S.A. Limits—Holes: Metric System

Quality 9 HOLES Limits in Microns (μ = 0·001 mm)

Nominal Size, mm	A9	B9	C9	D9	E9	F9	H9	J9	N9
1·01– 3	+ 295	+ 165	+ 85	+ 45	+ 39	+ 32	+ 25	+ 12	− 0
	+ 270	+ 140	+ 60	+ 20	+ 14	+ 7	+ 0	− 13	− 25
3·01– 6	+ 300	+ 170	+ 100	+ 60	+ 50	+ 40	+ 30	+ 15	− 0
	+ 270	+ 140	+ 70	+ 30	+ 20	+ 10	+ 0	− 15	− 30
6·01– 10	+ 316	+ 186	+ 116	+ 76	+ 61	+ 49	+ 36	+ 18	− 0
	+ 280	+ 150	+ 80	+ 40	+ 25	+ 13	+ 0	− 18	− 36
10·01– 14	+ 333	+ 193	+ 138	+ 93	+ 75	+ 59	+ 43	+ 21	− 0
	+ 290	+ 150	+ 95	+ 50	+ 32	+ 16	+ 0	− 22	− 43
14·01– 18	+ 333	+ 193	+ 138	+ 93	+ 75	+ 59	+ 43	+ 21	− 0
	+ 290	+ 150	+ 95	+ 50	+ 32	+ 16	+ 0	− 22	− 43
18·01– 24	+ 352	+ 212	+ 162	+ 117	+ 92	+ 72	+ 52	+ 26	− 0
	+ 300	+ 160	+ 110	+ 65	+ 40	+ 20	+ 0	− 26	− 52
24·01– 30	+ 352	+ 212	+ 162	+ 117	+ 92	+ 72	+ 52	+ 26	− 0
	+ 300	+ 160	+ 110	+ 65	+ 40	+ 20	+ 0	− 26	− 52
30·01– 40	+ 372	+ 232	+ 182	+ 142	+ 112	—	+ 62	+ 31	− 0
	+ 310	+ 170	+ 120	+ 80	+ 50	—	+ 0	− 31	− 62
40·01– 50	+ 382	+ 242	+ 192	+ 142	+ 112	—	+ 62	+ 31	− 0
	+ 320	+ 180	+ 130	+ 80	+ 50	—	+ 0	− 31	− 62
50·01– 65	+ 414	+ 264	+ 214	+ 174	+ 134	—	+ 74	+ 37	− 0
	+ 340	+ 190	+ 140	+ 100	+ 60	—	+ 0	− 37	− 74
65·01– 80	+ 434	+ 274	+ 224	+ 174	+ 134	—	+ 74	+ 37	− 0
	+ 360	+ 200	+ 150	+ 100	+ 60	—	+ 0	− 37	− 74
80·01–100	+ 467	+ 307	+ 257	+ 207	+ 159	—	+ 87	+ 43	− 0
	+ 380	+ 220	+ 170	+ 120	+ 72	—	+ 0	− 44	− 87
100·1 –120	+ 497	+ 327	+ 267	+ 207	+ 159	—	+ 87	+ 43	− 0
	+ 410	+ 240	+ 180	+ 120	+ 72	—	+ 0	− 44	− 87
120·1 –140	+ 560	+ 360	+ 300	+ 245	+ 185	—	+ 100	+ 50	− 0
	+ 460	+ 260	+ 200	+ 145	+ 85	—	+ 0	− 50	− 100
140·1 –160	+ 620	+ 380	+ 310	+ 245	+ 185	—	+ 100	+ 50	− 0
	+ 520	+ 280	+ 210	+ 145	+ 85	—	+ 0	− 50	− 100
160·1 –180	+ 680	+ 410	+ 330	+ 245	+ 185	—	+ 100	+ 50	− 0
	+ 580	+ 310	+ 230	+ 145	+ 85	—	+ 0	− 50	− 100
180·1 –200	+ 775	+ 455	+ 355	+ 285	+ 215	—	+ 115	+ 57	− 0
	+ 660	+ 340	+ 240	+ 170	+ 100	—	+ 0	− 58	− 115
200·1 –225	+ 855	+ 495	+ 375	+ 285	+ 215	—	+ 115	+ 57	− 0
	+ 740	+ 380	+ 260	+ 170	+ 100	—	+ 0	− 58	− 115
225·1 –250	+ 935	+ 535	+ 395	+ 285	+ 215	—	+ 115	+ 57	− 0
	+ 820	+ 420	+ 280	+ 170	+ 100	—	+ 0	− 58	− 115
250·1 –280	+ 1050	+ 610	+ 430	+ 320	+ 240	—	+ 130	+ 65	− 0
	+ 920	+ 480	+ 300	+ 190	+ 110	—	+ 0	− 65	− 130
280·1 –315	+ 1180	+ 670	+ 460	+ 320	+ 240	—	+ 130	+ 65	− 0
	+ 1050	+ 540	+ 330	+ 190	+ 110	—	+ 0	− 65	− 130
315·1 –355	+ 1340	+ 740	+ 500	+ 350	+ 265	—	+ 140	+ 70	− 0
	+ 1200	+ 600	+ 360	+ 210	+ 125	—	+ 0	− 70	− 140
355·1 –400	+ 1490	+ 820	+ 540	+ 350	+ 265	—	+ 140	+ 70	− 0
	+ 1350	+ 680	+ 400	+ 210	+ 125	—	+ 0	− 70	− 140
400·1 –450	+ 1655	+ 915	+ 595	+ 385	+ 290	—	+ 155	+ 77	− 0
	+ 1500	+ 760	+ 440	+ 230	+ 135	—	+ 0	− 78	− 155
450·1 –500	+ 1805	+ 995	+ 635	+ 385	+ 290	—	+ 155	+ 77	− 0
	+ 1650	+ 840	+ 480	+ 230	+ 135	—	+ 0	− 78	− 155

TABLE 21 (*contd.*)

I.S.A. Limits—Holes: Metric System

Qualities 10 and 11 HOLES Limits in Microns (μ = 0·001 mm)

Nominal Size, mm	D10	H10	J10	N10	A11	B11	C11	D11	H11	J11	N11
1·01– 3	+60	+40	+20	−0	+330	+200	+120	+80	+60	+30	−0
	+20	+0	−20	−40	+270	+140	+60	+20	+0	−30	−60
3·01– 6	+78	+48	+24	−0	+345	+215	+145	+105	+75	+37	−0
	+30	+0	−24	−48	+270	+140	+70	+30	+0	−38	−75
6·01– 10	+98	+58	+29	−0	+370	+240	+170	+130	+90	+45	−0
	+40	+0	−29	−58	+280	+150	+80	+40	+0	−45	−90
10·01– 14	+120	+70	+35	−0	+400	+260	+205	+160	+110	+55	−0
	+50	+0	−35	−70	+290	+150	+95	+50	+0	−55	−110
14·01– 18	+120	+70	+35	−0	+400	+260	+205	+160	+110	+55	−0
	+50	+0	−35	−70	+290	+150	+95	+50	+0	−55	−110
18·01– 24	+149	+84	+42	−0	+430	+290	+240	+195	+130	+65	−0
	+65	+0	−42	−84	+300	+160	+110	+65	+0	−65	−130
24·01– 30	+149	+84	+42	−0	+430	+290	+240	+195	+130	+65	−0
	+65	+0	−42	−84	+300	+160	+110	+65	+0	−65	−130
30·01– 40	+180	+100	+50	−0	+470	+330	+280	+240	+160	+80	−0
	+80	+0	−50	−100	+310	+170	+120	+80	+0	−80	−160
40·01– 50	+180	+100	+50	−0	+480	+340	+290	+240	+160	+80	−0
	+80	+0	−50	−100	+320	+180	+130	+80	+0	−80	−160
50·01– 65	+220	+120	+60	−0	+530	+380	+330	+290	+190	+95	−0
	+100	+0	−60	−120	+340	+190	+140	+100	+0	−95	−190
65·01– 80	+220	+120	+60	−0	+550	+390	+340	+290	+190	+95	−0
	+100	+0	−60	−120	+360	+200	+150	+100	+0	−95	−190
80·01–100	+260	+140	+70	−0	+600	+440	+390	+340	+220	+110	−0
	+120	+0	−70	−140	+380	+220	+170	+120	+0	−110	−220
100·1 –120	+260	+140	+70	−0	+630	+460	+400	+340	+220	+110	−0
	+120	+0	−70	−140	+410	+240	+180	+120	+0	−110	−220
120·1 –140	+305	+160	+80	−0	+710	+510	+450	+395	+250	+125	−0
	+145	+0	−80	−160	+460	+260	+200	+145	+0	−125	−250
140·1 –160	+305	+160	+80	−0	+770	+530	+460	+395	+250	+125	−0
	+145	+0	−80	−160	+520	+280	+210	+145	+0	−125	−250
160·1 –180	+305	+160	+80	−0	+830	+560	+480	+395	+250	+125	−0
	+145	+0	−80	−160	+580	+310	+230	+145	+0	−125	−250
180·1 –200	+355	+185	+92	−0	+950	+630	+530	+460	+290	+145	−0
	+170	+0	−93	−185	+660	+340	+240	+170	+0	−145	−290
200·1 –225	+355	+185	+92	−0	+1030	+670	+550	+460	+290	+145	−0
	+170	+0	−93	−185	+740	+380	+260	+170	+0	−145	−290
225·1 –250	+355	+185	+92	−0	+1110	+710	+570	+460	+290	+145	−0
	+170	+0	−93	−185	+820	+420	+280	+170	+0	−145	−290
250·1 –280	+400	+210	+105	−0	+1240	+800	+620	+510	+320	+160	−0
	+190	+0	−105	−210	+920	+480	+300	+190	+0	−160	−320
280·1 –315	+400	+210	+105	−0	+1370	+860	+650	+510	+320	+160	−0
	+190	+0	−105	−210	+1050	+540	+330	+190	+0	−160	−320
315·1 –355	+440	+230	+115	−0	+1560	+960	+720	+570	+360	+180	−0
	+210	+0	−115	−230	+1200	+600	+360	+210	+0	−180	−360
355·1 –400	+440	+230	+115	−0	+1710	+1040	+760	+570	+360	+180	−0
	+210	+0	−115	−230	+1350	+680	+400	+210	+0	−180	−360
400·1 –450	+480	+250	+125	−0	+1900	+1160	+840	+630	+400	+200	−0
	+230	+0	−125	−250	+1500	+760	+440	+230	+0	−200	−400
450·1 –500	+480	+250	+125	−0	+2050	+1240	+880	+630	+400	+200	−0
	+230	+0	−125	−250	+1650	+840	+480	+230	+0	−200	−400

TABLE 22

I.S.A. Limits—Shafts: Metric System

Quality 5 SHAFTS Limits in Microns (μ = 0·001 mm)

Nominal Size, mm	g5	h5	j5	k5	m5	n5	p5	r5	s5	t5	u5	(v5)	x5
1·01– 3	−3	−0	+4	—	+7	+11	+14	+17	+20	—	+23	—	+27
	−8	−5	−1	—	+2	+6	+9	+12	+15	—	+18	—	+22
3·01– 6	−4	−0	+4	—	+9	+13	+17	+20	+24	—	+28	—	+33
	−9	−5	−1	—	+4	+8	+12	+15	+19	—	+23	—	+28
6·01– 10	−5	−0	+4	+7	+12	+16	+21	+25	+29	—	+34	—	+40
	−11	−6	−2	+1	+6	+10	+15	+19	+23	—	+28	—	+34
10·01– 14	−6	−0	+5	+9	+15	+20	+26	+31	+36	—	+41	—	+48
	−14	−8	−3	+1	+7	+12	+18	+23	+28	—	+33	—	+40
14·01– 18	−6	−0	+5	+9	+15	+20	+26	+31	+36	—	+41	+47	+53
	−14	−8	−3	+1	+7	+12	+18	+23	+28	—	+33	+39	+45
18·01– 24	−7	−0	+5	+11	+17	+24	+31	+37	+44	—	+50	+56	+63
	−16	−9	−4	+2	+8	+15	+22	+28	+35	—	+41	+47	+54
24·01– 30	−7	−0	+5	+11	+17	+24	+31	+37	+44	+50	+57	+64	+73
	−16	−9	−4	+2	+8	+15	+22	+28	+35	+41	+48	+55	+64
30·01– 40	−9	−0	+6	+13	+20	+28	+37	+45	+54	+59	+71	+79	+91
	−20	−11	−5	+2	+9	+17	+26	+34	+43	+48	+60	+68	+80
40·01– 50	−9	−0	+6	+13	+20	+28	+37	+45	+54	+65	+81	+92	+108
	−20	−11	−5	+2	+9	+17	+26	+34	+43	+54	+70	+81	+97
50·01– 65	−10	−0	+6	+15	+24	+33	+45	+54	+66	+79	+100	+115	+135
	−23	−13	−7	+2	+11	+20	+32	+41	+53	+66	+87	+102	+122
65·01– 80	−10	−0	+6	+15	+24	+33	+45	+56	+72	+88	+115	+133	+159
	−23	−13	−7	+2	+11	+20	+32	+43	+59	+75	+102	+120	+146
80·01–100	−12	−0	+6	+18	+28	+38	+52	+66	+86	+106	+139	+161	+193
	−27	−15	−9	+3	+13	+23	+37	+51	+71	+91	+124	+146	+178
100·1 –120	−12	−0	+6	+18	+28	+38	+52	+69	+94	+119	+159	+187	+225
	−27	−15	−9	+3	+13	+23	+37	+54	+79	+104	+144	+172	+210
120·1 –140	−14	−0	+7	+21	+33	+45	+61	+81	+110	+140	+188	+220	+266
	−32	−18	−11	+3	+15	+27	+43	+63	+92	+122	+170	+202	+248
140·1 –160	−14	−0	+7	+21	+33	+45	+61	+83	+118	+152	+208	+246	+298
	−32	−18	−11	+3	+15	+27	+43	+65	+100	+134	+190	+228	+280
160·1 –180	−14	−0	+7	+21	+33	+45	+61	+86	+126	+164	+228	+270	+328
	−32	−18	−11	+3	+15	+27	+43	+68	+108	+146	+210	+252	+310
180·1 –200	−15	−0	+7	+24	+37	+51	+70	+97	+142	+186	+256	+304	+370
	−35	−20	−13	+4	+17	+31	+50	+77	+122	+166	+236	+284	+350
200·1 –225	−15	−0	+7	+24	+37	+51	+70	+100	+150	+200	+278	+330	+405
	−35	−20	−13	+4	+17	+31	+50	+80	+130	+180	+258	+310	+385
225·1 –250	−15	−0	+7	+24	+37	+51	+70	+104	+160	+216	+304	+360	+445
	−35	−20	−13	+4	+17	+31	+50	+84	+140	+196	+284	+340	+425
250·1 –280	−17	−0	+7	+27	+43	+57	+79	+117	+181	+241	+338	+408	+498
	−40	−23	−16	+4	+20	+34	+56	+94	+158	+218	+315	+385	+475
280·1 –315	−17	−0	+7	+27	+43	+57	+79	+121	+193	+263	+373	+448	+548
	−40	−23	−16	+4	+20	+34	+56	+98	+170	+240	+350	+425	+525
315·1 –355	−18	−0	+7	+29	+46	+62	+87	+133	+215	+293	+415	+500	+615
	−43	−25	−18	+4	+21	+37	+62	+108	+190	+268	+390	+475	+590
355·1 –400	−18	−0	+7	+29	+46	+62	+87	+139	+233	+319	+460	+555	+685
	−43	−25	−18	+4	+21	+37	+62	+114	+208	+294	+435	+530	+660
400·1 –450	−20	−0	+7	+32	+50	+67	+95	+153	+259	+357	+517	+622	+767
	−47	−27	−20	+5	+23	+40	+68	+126	+232	+330	+490	+595	+740
450·1 –500	−20	−0	+7	+32	+50	+67	+95	+159	+279	+387	+567	+687	+847
	−47	−27	−20	+5	+23	+40	+68	+132	+252	+360	+540	+660	+820

TABLE 22 (*contd.*). I.S.A. LIMITS—

Quality 6 SHAFTS Limits in Microns

Nominal Size, mm	f6	g6	h6	j6	k6	m6	n6	p6
1·01– 3	− 7	− 3	− 0	+ 6	—	+ 9	+ 13	+ 16
	− 14	− 10	− 7	− 1	—	+ 2	+ 6	+ 9
3·01– 6	− 10	− 4	− 0	+ 7	—	+ 12	+ 16	+ 20
	− 18	− 12	− 8	− 1	—	+ 4	+ 8	+ 12
6·01– 10	− 13	− 5	− 0	+ 7	+ 10	+ 15	+ 19	+ 24
	− 22	− 14	− 9	− 2	+ 1	+ 6	+ 10	+ 15
10·01– 14	− 16	− 6	− 0	+ 8	+ 12	+ 18	+ 23	+ 29
	− 27	− 17	− 11	− 3	+ 1	+ 7	+ 12	+ 18
14·01– 18	− 16	− 6	− 0	+ 8	+ 12	+ 18	+ 23	+ 29
	− 27	− 17	− 11	− 3	+ 1	+ 7	+ 12	+ 18
18·01– 24	− 20	− 7	− 0	+ 9	+ 15	+ 21	+ 28	+ 35
	− 33	− 20	− 13	− 4	+ 2	+ 8	+ 15	+ 22
24·01– 30	− 20	− 7	− 0	+ 9	+ 15	+ 21	+ 28	+ 35
	− 33	− 20	− 13	− 4	+ 2	+ 8	+ 15	+ 22
30·01– 40	− 25	− 9	− 0	+ 11	+ 18	+ 25	+ 33	+ 42
	− 41	− 25	− 16	− 5	+ 2	+ 9	+ 17	+ 26
40·01– 50	− 25	− 9	− 0	+ 11	+ 18	+ 25	+ 33	+ 42
	− 41	− 25	− 16	− 5	+ 2	+ 9	+ 17	+ 26
50·01– 65	− 30	− 10	− 0	+ 12	+ 21	+ 30	+ 39	+ 51
	− 49	− 29	− 19	− 7	+ 2	+ 11	+ 20	+ 32
65·01– 80	− 30	− 10	− 0	+ 12	+ 21	+ 30	+ 39	+ 51
	− 49	− 29	− 19	− 7	+ 2	+ 11	+ 20	+ 32
80·01–100	− 36	− 12	− 0	+ 13	+ 25	+ 35	+ 45	+ 59
	− 58	− 34	− 22	− 9	+ 3	+ 13	+ 23	+ 37
100·1 –120	− 36	− 12	− 0	+ 13	+ 25	+ 35	+ 45	+ 59
	− 58	− 34	− 22	− 9	+ 3	+ 13	+ 23	+ 37
120·1 –140	− 43	− 14	− 0	+ 14	+ 28	+ 40	+ 52	+ 68
	− 68	− 39	− 25	− 11	+ 3	+ 15	+ 27	+ 43
140·1 –160	− 43	− 14	− 0	+ 14	+ 28	+ 40	+ 52	+ 68
	− 68	− 39	− 25	− 11	+ 3	+ 15	+ 27	+ 43
160·1 –180	− 43	− 14	− 0	+ 14	+ 28	+ 40	+ 52	+ 68
	− 68	− 39	− 25	− 11	+ 3	+ 15	+ 27	+ 43
180·1 –200	− 50	− 15	− 0	+ 16	+ 33	+ 46	+ 60	+ 79
	− 79	− 44	− 29	− 13	+ 4	+ 17	+ 31	+ 50
200·1 –225	− 50	− 15	− 0	+ 16	+ 33	+ 46	+ 60	+ 79
	− 79	− 44	− 29	− 13	+ 4	+ 17	+ 31	+ 50
225·1 –250	− 50	− 15	− 0	+ 16	+ 33	+ 46	+ 60	+ 79
	− 79	− 44	− 29	− 13	+ 4	+ 17	+ 31	+ 50
250·1 –280	− 56	− 17	− 0	+ 16	+ 36	+ 52	+ 66	+ 88
	− 88	− 49	− 32	− 16	+ 4	+ 20	+ 34	+ 56
280·1 –315	− 56	− 17	− 0	+ 16	+ 36	+ 52	+ 66	+ 88
	− 88	− 49	− 32	− 16	+ 4	+ 20	+ 34	+ 56
315·1 –355	− 62	− 18	− 0	+ 18	+ 40	+ 57	+ 73	+ 98
	− 98	− 54	− 36	− 18	+ 4	+ 21	+ 37	+ 62
355·1 –400	− 62	− 18	− 0	+ 18	+ 40	+ 57	+ 73	+ 98
	− 98	− 54	− 36	− 18	+ 4	+ 21	+ 37	+ 62
400·1 –450	− 68	− 20	− 0	+ 20	+ 45	+ 63	+ 80	+ 108
	− 108	− 60	− 40	− 20	+ 5	+ 23	+ 40	+ 68
450·1 –500	− 68	− 20	− 0	+ 20	+ 45	+ 63	+ 80	+ 108
	− 108	− 60	− 40	− 20	+ 5	+ 23	+ 40	+ 68

Shafts: Metric System
(μ = 0·001 mm)

r6	s6	t6	u6	(v6)	x6	(y6)	z6
+ 19	+ 22	—	+ 25	—	+ 29	—	+ 35
+ 12	+ 15	—	+ 18	—	+ 22	—	+ 28
+ 23	+ 27	—	+ 31	—	+ 36	—	+ 43
+ 15	+ 19	—	+ 23	—	+ 28	—	+ 35
+ 28	+ 32	—	+ 37	—	+ 43	—	+ 51
+ 19	+ 23	—	+ 28	—	+ 34	—	+ 42
+ 34	+ 39	—	+ 44	—	+ 51	—	+ 61
+ 23	+ 28	—	+ 33	—	+ 40	—	+ 50
+ 34	+ 39	—	+ 44	+ 50	+ 56	—	+ 71
+ 23	+ 28	—	+ 33	+ 39	+ 45	—	+ 60
+ 41	+ 48	—	+ 54	+ 60	+ 67	+ 76	+ 86
+ 28	+ 35	—	+ 41	+ 47	+ 54	+ 63	+ 73
+ 41	+ 48	+ 54	+ 61	+ 68	+ 77	+ 88	+ 101
+ 28	+ 35	+ 41	+ 48	+ 55	+ 64	+ 75	+ 88
+ 50	+ 59	+ 64	+ 76	+ 84	+ 96	+ 110	+ 128
+ 34	+ 43	+ 48	+ 60	+ 68	+ 80	+ 94	+ 112
+ 50	+ 59	+ 70	+ 86	+ 97	+ 113	+ 130	+ 152
+ 34	+ 43	+ 54	+ 70	+ 81	+ 97	+ 114	+ 136
+ 60	+ 72	+ 85	+ 106	+ 121	+ 141	+ 163	+ 191
+ 41	+ 53	+ 66	+ 87	+ 102	+ 122	+ 144	+ 172
+ 62	+ 78	+ 94	+ 121	+ 139	+ 165	+ 193	+ 229
+ 43	+ 59	+ 75	+ 102	+ 120	+ 146	+ 174	+ 210
+ 73	+ 93	+ 113	+ 146	+ 168	+ 200	+ 236	+ 280
+ 51	+ 71	+ 91	+ 124	+ 146	+ 178	+ 214	+ 258
+ 76	+ 101	+ 126	+ 166	+ 194	+ 232	+ 276	+ 332
+ 54	+ 79	+ 104	+ 144	+ 172	+ 210	+ 254	+ 310
+ 88	+ 117	+ 147	+ 195	+ 227	+ 273	+ 325	+ 390
+ 63	+ 92	+ 122	+ 170	+ 202	+ 248	+ 300	+ 365
+ 90	+ 125	+ 159	+ 215	+ 253	+ 305	+ 365	+ 440
+ 65	+ 100	+ 134	+ 190	+ 228	+ 280	+ 340	+ 415
+ 93	+ 133	+ 171	+ 235	+ 277	+ 335	+ 405	+ 490
+ 68	+ 108	+ 146	+ 210	+ 252	+ 310	+ 380	+ 465
+ 106	+ 151	+ 195	+ 265	+ 313	+ 379	+ 454	+ 549
+ 77	+ 122	+ 166	+ 236	+ 284	+ 350	+ 425	+ 520
+ 109	+ 159	+ 209	+ 287	+ 339	+ 414	+ 499	+ 604
+ 80	+ 130	+ 180	+ 258	+ 310	+ 385	+ 470	+ 575
+ 113	+ 169	+ 225	+ 313	+ 369	+ 454	+ 549	+ 669
+ 84	+ 140	+ 196	+ 284	+ 340	+ 425	+ 520	+ 640
+ 126	+ 190	+ 250	+ 347	+ 417	+ 507	+ 612	+ 742
+ 94	+ 158	+ 218	+ 315	+ 385	+ 475	+ 580	+ 710
+ 130	+ 202	+ 272	+ 382	+ 457	+ 557	+ 682	+ 822
+ 98	+ 170	+ 240	+ 350	+ 425	+ 525	+ 650	+ 790
+ 144	+ 226	+ 304	+ 426	+ 511	+ 626	+ 766	+ 936
+ 108	+ 190	+ 268	+ 390	+ 475	+ 590	+ 730	+ 900
+ 150	+ 244	+ 330	+ 471	+ 566	+ 696	+ 856	+ 1036
+ 114	+ 208	+ 294	+ 435	+ 530	+ 660	+ 820	+ 1000
+ 166	+ 272	+ 370	+ 530	+ 635	+ 780	+ 960	+ 1140
+ 126	+ 232	+ 330	+ 490	+ 595	+ 740	+ 920	+ 1100
+ 172	+ 292	+ 400	+ 580	+ 700	+ 860	+ 1040	+ 1290
+ 132	+ 252	+ 360	+ 540	+ 660	+ 820	+ 1000	+ 1250

TABLE 22 (*contd.*). I.S.A. LIMITS—

Quality 7 SHAFTS Limits in Microns

Nominal Size, mm	e7	f7	h7	j7	k7	m7	n7	p7
1·01– 3	− 14	− 7	− 0	+ 7	—	—	+ 15	+ 18
	− 23	− 16	− 9	− 2	—	—	+ 6	+ 9
3·01– 6	− 20	− 10	− 0	+ 9	—	—	+ 20	+ 24
	− 32	− 22	− 12	− 3	—	—	+ 8	+ 12
6·01– 10	− 25	− 13	− 0	+ 10	+ 16	+ 21	+ 25	+ 30
	− 40	− 28	− 15	− 5	+ 1	+ 6	+ 10	+ 15
10·01– 14	− 32	− 16	− 0	+ 12	+ 19	+ 25	+ 30	+ 36
	− 50	− 34	− 18	− 6	+ 1	+ 7	+ 12	+ 18
14·01– 18	− 32	− 16	− 0	+ 12	+ 19	+ 25	+ 30	+ 36
	− 50	− 34	− 18	− 6	+ 1	+ 7	+ 12	+ 18
18·01– 24	− 40	− 20	− 0	+ 13	+ 23	+ 29	+ 36	+ 43
	− 61	− 41	− 21	− 8	+ 2	+ 8	+ 15	+ 22
24·01– 30	− 40	− 20	− 0	+ 13	+ 23	+ 29	+ 36	+ 43
	− 61	− 41	− 21	− 8	+ 2	+ 8	+ 15	+ 22
30·01– 40	− 50	− 25	− 0	+ 15	+ 27	+ 34	+ 42	+ 51
	− 75	− 50	− 25	− 10	+ 2	+ 9	+ 17	+ 26
40·01– 50	− 50	− 25	− 0	+ 15	+ 27	+ 34	+ 42	+ 51
	− 75	− 50	− 25	− 10	+ 2	+ 9	+ 17	+ 26
50·01– 65	− 60	− 30	− 0	+ 18	+ 32	+ 41	+ 50	+ 62
	− 90	− 60	− 30	− 12	+ 2	+ 11	+ 20	+ 32
65·01– 80	− 60	− 30	− 0	+ 18	+ 32	+ 41	+ 50	+ 62
	− 90	− 60	− 30	− 12	+ 2	+ 11	+ 20	+ 32
80·01–100	− 72	− 36	− 0	+ 20	+ 38	+ 48	+ 58	+ 72
	− 107	− 71	− 35	− 15	+ 3	+ 13	+ 23	+ 37
100·1 –120	− 72	− 36	− 0	+ 20	+ 38	+ 48	+ 58	+ 72
	− 107	− 71	− 35	− 15	+ 3	+ 13	+ 23	+ 37
120·1 –140	− 85	− 43	− 0	+ 22	+ 43	+ 55	+ 67	+ 83
	− 125	− 83	− 40	− 18	+ 3	+ 15	+ 27	+ 43
140·1 –160	− 85	− 43	− 0	+ 22	+ 43	+ 55	+ 67	+ 83
	− 125	− 83	− 40	− 18	+ 3	+ 15	+ 27	+ 43
160·1 –180	− 85	− 43	− 0	+ 22	+ 43	+ 55	+ 67	+ 83
	− 125	− 83	− 40	− 18	+ 3	+ 15	+ 27	+ 43
180·1 –200	− 100	− 50	− 0	+ 25	+ 50	+ 63	+ 77	+ 96
	− 146	− 96	− 46	− 21	+ 4	+ 17	+ 31	+ 50
200·1 –225	− 100	− 50	− 0	+ 25	+ 50	+ 63	+ 77	+ 96
	− 146	− 96	− 46	− 21	+ 4	+ 17	+ 31	+ 50
225·1 –250	− 100	− 50	− 0	+ 25	+ 50	+ 63	+ 77	+ 96
	− 146	− 96	− 46	− 21	+ 4	+ 17	+ 31	+ 50
250·1 –280	− 110	− 56	− 0	+ 26	+ 56	+ 72	+ 86	+ 108
	− 162	− 108	− 52	− 26	+ 4	+ 20	+ 34	+ 56
280·1 –315	− 110	− 56	− 0	+ 26	+ 56	+ 72	+ 86	+ 108
	− 162	− 108	− 52	− 26	+ 4	+ 20	+ 34	+ 56
315·1 –355	− 125	− 62	− 0	+ 29	+ 61	+ 78	+ 94	+ 119
	− 182	− 119	− 57	− 28	+ 4	+ 21	+ 37	+ 62
355·1 –400	− 125	− 62	− 0	+ 29	+ 61	+ 78	+ 94	+ 119
	− 182	− 119	− 57	− 28	+ 4	+ 21	+ 37	+ 62
400·1 –450	− 135	− 68	− 0	+ 31	+ 68	+ 86	+ 103	+ 131
	− 198	− 131	− 63	− 32	+ 5	+ 23	+ 40	+ 68
450·1 –500	− 135	− 68	− 0	+ 31	+ 68	+ 86	+ 103	+ 131
	− 198	− 131	− 63	− 32	+ 5	+ 23	+ 40	+ 68

Shafts: Metric System

($\mu = 0{\cdot}001$ mm)

r7	s7	t7	u7	(v7)	x7	(y7)	z7
+ 21	+ 24	—	+ 27	—	+ 31	—	+ 37
+ 12	+ 15	—	+ 18	—	+ 22	—	+ 28
+ 27	+ 31	—	+ 35	—	+ 40	—	+ 47
+ 15	+ 19	—	+ 23	—	+ 28	—	+ 35
+ 34	+ 38	—	+ 43	—	+ 49	—	+ 57
+ 19	+ 23	—	+ 28	—	+ 34	—	+ 42
+ 41	+ 46	—	+ 51	—	+ 58	—	+ 68
+ 23	+ 28	—	+ 33	—	+ 40	—	+ 50
+ 41	+ 46	—	+ 51	+ 57	+ 63	—	+ 78
+ 23	+ 28	—	+ 33	+ 39	+ 45	—	+ 60
+ 49	+ 56	—	+ 62	+ 68	+ 75	+ 84	+ 94
+ 28	+ 35	—	+ 41	+ 47	+ 54	+ 63	+ 73
+ 49	+ 56	+ 62	+ 69	+ 76	+ 85	+ 96	+ 109
+ 28	+ 35	+ 41	+ 48	+ 55	+ 64	+ 75	+ 88
+ 59	+ 68	+ 73	+ 85	+ 93	+ 105	+ 119	+ 137
+ 34	+ 43	+ 48	+ 60	+ 68	+ 80	+ 94	+ 112
+ 59	+ 68	+ 79	+ 95	+ 106	+ 122	+ 139	+ 161
+ 34	+ 43	+ 54	+ 70	+ 81	+ 97	+ 114	+ 136
+ 71	+ 83	+ 96	+ 117	+ 132	+ 152	+ 174	+ 202
+ 41	+ 53	+ 66	+ 87	+ 102	+ 122	+ 144	+ 172
+ 73	+ 89	+ 105	+ 132	+ 150	+ 176	+ 204	+ 240
+ 43	+ 59	+ 75	+ 102	+ 120	+ 146	+ 174	+ 210
+ 86	+ 106	+ 126	+ 159	+ 181	+ 213	+ 249	+ 293
+ 51	+ 71	+ 91	+ 124	+ 146	+ 178	+ 214	+ 258
+ 89	+ 114	+ 139	+ 179	+ 207	+ 245	+ 289	+ 345
+ 54	+ 79	+ 104	+ 144	+ 172	+ 210	+ 254	+ 310
+ 103	+ 132	+ 162	+ 210	+ 242	+ 288	+ 340	+ 405
+ 63	+ 92	+ 122	+ 170	+ 202	+ 248	+ 300	+ 365
+ 105	+ 140	+ 174	+ 230	+ 268	+ 320	+ 380	+ 455
+ 65	+ 100	+ 134	+ 190	+ 228	+ 280	+ 340	+ 415
+ 108	+ 148	+ 186	+ 250	+ 292	+ 350	+ 420	+ 505
+ 68	+ 108	+ 146	+ 210	+ 252	+ 310	+ 380	+ 465
+ 123	+ 168	+ 212	+ 282	+ 330	+ 396	+ 471	+ 566
+ 77	+ 122	+ 166	+ 236	+ 284	+ 350	+ 425	+ 520
+ 126	+ 176	+ 226	+ 304	+ 356	+ 431	+ 516	+ 621
+ 80	+ 130	+ 180	+ 258	+ 310	+ 385	+ 470	+ 575
+ 130	+ 186	+ 242	+ 330	+ 386	+ 471	+ 566	+ 686
+ 84	+ 140	+ 196	+ 284	+ 340	+ 425	+ 520	+ 640
+ 146	+ 210	+ 270	+ 367	+ 437	+ 527	+ 632	+ 762
+ 94	+ 158	+ 218	+ 315	+ 385	+ 475	+ 580	+ 710
+ 150	+ 222	+ 292	+ 402	+ 477	+ 577	+ 702	+ 842
+ 98	+ 170	+ 240	+ 350	+ 425	+ 525	+ 650	+ 790
+ 165	+ 247	+ 325	+ 447	+ 532	+ 647	+ 787	+ 957
+ 108	+ 190	+ 268	+ 390	+ 475	+ 590	+ 730	+ 900
+ 171	+ 265	+ 351	+ 492	+ 587	+ 717	+ 877	+ 1057
+ 114	+ 208	+ 294	+ 435	+ 530	+ 660	+ 820	+ 1000
+ 189	+ 295	+ 393	+ 553	+ 658	+ 803	+ 983	+ 1163
+ 126	+ 232	+ 330	+ 490	+ 595	+ 740	+ 920	+ 1100
+ 195	+ 315	+ 423	+ 603	+ 723	+ 883	+ 1063	+ 1313
+ 132	+ 252	+ 360	+ 540	+ 660	+ 820	+ 1000	+ 1250

TABLE 22 (*contd.*). I.S.A. LIMITS—

Qualities 8 and 9 — SHAFTS — Limits in Microns

Nominal Size, mm	b8	c8	d8	e8	f8	h8	j8	k8
1·01– 3	− 140 − 154	− 60 − 74	− 20 − 34	− 14 − 28	− 7 − 21	− 0 − 14	+ 7 − 7	+ 14 + 0
3·01– 6	− 140 − 158	− 70 − 88	− 30 − 48	− 20 − 38	− 10 − 28	− 0 − 18	+ 9 − 9	+ 18 + 0
6·01– 10	− 150 − 172	− 80 − 102	− 40 − 62	− 25 − 47	− 13 − 35	− 0 − 22	+ 11 − 11	+ 22 + 0
10·01– 14	− 150 − 177	− 95 − 122	− 50 − 77	− 32 − 59	− 16 − 43	− 0 − 27	+ 14 − 13	+ 27 + 0
14·01– 18	− 150 − 177	− 95 − 122	− 50 − 77	− 32 − 59	− 16 − 43	− 0 − 27	+ 14 − 13	+ 27 + 0
18·01– 24	− 160 − 193	− 110 − 143	− 65 − 98	− 40 − 73	− 20 − 53	− 0 − 33	+ 17 − 16	+ 33 + 0
24·01– 30	− 160 − 193	− 110 − 143	− 65 − 98	− 40 − 73	− 20 − 53	− 0 − 33	+ 17 − 16	+ 33 + 0
30·01– 40	− 170 − 209	− 120 − 159	− 80 − 119	− 50 − 89	− 25 − 64	− 0 − 39	+ 20 − 19	+ 39 + 0
40·01– 50	− 180 − 219	− 130 − 169	− 80 − 119	− 50 − 89	− 25 − 64	− 0 − 39	+ 20 − 19	+ 39 + 0
50·01– 65	− 190 − 236	− 140 − 186	− 100 − 146	− 60 − 106	− 30 − 76	− 0 − 46	+ 23 − 23	+ 46 + 0
65·01– 80	− 200 − 246	− 150 − 196	− 100 − 146	− 60 − 106	− 30 − 76	− 0 − 46	+ 23 − 23	+ 46 + 0
80·01–100	− 220 − 274	− 170 − 224	− 120 − 174	− 72 − 126	− 36 − 90	− 0 − 54	+ 27 − 27	+ 54 + 0
100·1 –120	− 240 − 294	− 180 − 234	− 120 − 174	− 72 − 126	− 36 − 90	− 0 − 54	+ 27 − 27	+ 54 + 0
120·1 –140	− 260 − 323	− 200 − 263	− 145 − 208	− 85 − 148	− 43 − 106	− 0 − 63	+ 32 − 31	+ 63 + 0
140·1 –160	− 280 − 343	− 210 − 273	− 145 − 208	− 85 − 148	− 43 − 106	− 0 − 63	+ 32 − 31	+ 63 + 0
160·1 –180	− 310 − 373	− 230 − 293	− 145 − 208	− 85 − 148	− 43 − 106	− 0 − 63	+ 32 − 31	+ 63 + 0
180·1 –200	− 340 − 412	− 240 − 312	− 170 − 242	− 100 − 172	− 50 − 122	− 0 − 72	+ 36 − 36	+ 72 + 0
200·1 –225	− 380 − 452	− 260 − 332	− 170 − 242	− 100 − 172	− 50 − 122	− 0 − 72	+ 36 − 36	+ 72 + 0
225·1 –250	− 420 − 492	− 280 − 352	− 170 − 242	− 100 − 172	− 50 − 122	− 0 − 72	+ 36 − 36	+ 72 + 0
250·1 –280	− 480 − 561	− 300 − 381	− 190 − 271	− 110 − 191	− 56 − 137	− 0 − 81	+ 41 − 40	+ 81 + 0
280·1 –315	− 540 − 621	− 330 − 411	− 190 − 271	− 110 − 191	− 56 − 137	− 0 − 81	+ 41 − 40	+ 81 + 0
315·1 –355	− 600 − 689	− 360 − 449	− 210 − 299	− 125 − 214	− 62 − 151	− 0 − 89	+ 45 − 44	+ 89 + 0
355·1 –400	− 680 − 769	− 400 − 489	− 210 − 299	− 125 − 214	− 62 − 151	− 0 − 89	+ 45 − 44	+ 89 + 0
400·1 –450	− 760 − 857	− 440 − 537	− 230 − 327	− 135 − 232	− 68 − 165	− 0 − 97	+ 49 − 48	+ 97 + 0
450·1 –500	− 840 − 937	− 480 − 577	− 230 − 327	− 135 − 232	− 68 − 165	− 0 − 97	+ 49 − 48	+ 97 + 0

SHAFTS: METRIC SYSTEM
(μ = 0·001 mm)

a9	b9	c9	d9	e9	h9	j9	k9
− 270	− 140	− 60	− 20	− 14	− 0	+ 13	+ 25
− 295	− 165	− 85	− 45	− 39	− 25	− 12	+ 0
− 270	− 140	− 70	− 30	− 20	− 0	+ 15	+ 30
− 300	− 170	− 100	− 60	− 50	− 30	− 15	+ 0
− 280	− 150	− 80	− 40	− 25	− 0	+ 18	+ 36
− 316	− 186	− 116	− 76	− 61	− 36	− 18	+ 0
− 290	− 150	− 95	− 50	− 32	− 0	+ 22	+ 43
− 333	− 193	− 138	− 93	− 75	− 43	− 21	+ 0
− 290	− 150	− 95	− 50	− 32	− 0	+ 22	+ 43
− 333	− 193	− 138	− 93	− 75	− 43	− 21	+ 0
− 300	− 160	− 110	− 65	− 40	− 0	+ 26	+ 52
− 352	− 212	− 162	− 117	− 92	− 52	− 26	+ 0
− 300	− 160	− 110	− 65	− 40	− 0	+ 26	+ 52
− 352	− 212	− 162	− 117	− 92	− 52	− 26	+ 0
− 310	− 170	− 120	− 80	− 50	− 0	+ 31	+ 62
− 372	− 232	− 182	− 142	− 112	− 62	− 31	+ 0
− 320	− 180	− 130	− 80	− 50	− 0	+ 31	+ 62
− 382	− 242	− 192	− 142	− 112	− 62	− 31	+ 0
− 340	− 190	− 140	− 100	− 60	− 0	+ 37	+ 74
− 414	− 264	− 214	− 174	− 134	− 74	− 37	+ 0
− 360	− 200	− 150	− 100	− 60	− 0	+ 37	+ 74
− 434	− 274	− 224	− 174	− 134	− 74	− 37	+ 0
− 380	− 220	− 170	− 120	− 72	− 0	+ 44	+ 87
− 467	− 307	− 257	− 207	− 159	− 87	− 43	+ 0
− 410	− 240	− 180	− 120	− 72	− 0	+ 44	+ 87
− 497	− 327	− 267	− 207	− 159	− 87	− 43	+ 0
− 460	− 260	− 200	− 145	− 85	− 0	+ 50	+ 100
− 560	− 360	− 300	− 245	− 185	− 100	− 50	+ 0
− 520	− 280	− 210	− 145	− 85	− 0	+ 50	+ 100
− 620	− 380	− 310	− 245	− 185	− 100	− 50	+ 0
− 580	− 310	− 230	− 145	− 85	− 0	+ 50	+ 100
− 680	− 410	− 330	− 245	− 185	− 100	− 50	+ 0
− 660	− 340	− 240	− 170	− 100	− 0	+ 58	+ 115
− 775	− 455	− 355	− 285	− 215	− 115	− 57	+ 0
− 740	− 380	− 260	− 170	− 100	− 0	+ 58	+ 115
− 855	− 495	− 375	− 285	− 215	− 115	− 57	+ 0
− 820	− 420	− 280	− 170	− 100	− 0	+ 58	+ 115
− 935	− 535	− 395	− 285	− 215	− 115	− 57	+ 0
− 920	− 480	− 300	− 190	− 110	− 0	+ 65	+ 130
− 1050	− 610	− 430	− 320	− 240	− 130	− 65	+ 0
− 1050	− 540	− 330	− 190	− 110	− 0	+ 65	+ 130
− 1180	− 670	− 460	− 320	− 240	− 130	− 65	+ 0
− 1200	− 600	− 360	− 210	− 125	− 0	+ 70	+ 140
− 1340	− 740	− 500	− 350	− 265	− 140	− 70	+ 0
− 1350	− 680	− 400	− 210	− 125	− 0	+ 70	+ 140
− 1490	− 820	− 540	− 350	− 265	− 140	− 70	+ 0
− 1500	− 760	− 440	− 230	− 135	− 0	+ 78	+ 155
− 1655	− 915	− 595	− 385	− 290	− 155	− 77	+ 0
− 1650	− 840	− 480	− 230	− 135	− 0	+ 78	+ 155
− 1805	− 995	− 635	− 385	− 290	− 155	− 77	+ 0

TABLE 22 (*contd.*)

I.S.A. LIMITS—SHAFTS: METRIC SYSTEM

Qualities 10 and 11 SHAFTS Limits in Microns (μ = 0·001 mm)

Nominal Size, mm	d10	h10	j10	k10	a11	b11	c11	d11	h11	j11	k11
1·01– 3	− 20	− 0	+ 20	+ 40	− 270	− 140	− 60	− 20	− 0	+ 30	+ 60
	− 60	− 40	− 20	+ 0	− 330	− 200	−120	− 80	− 60	− 30	+ 0
3·01– 6	− 30	− 0	+ 24	+ 48	− 270	− 140	− 70	− 30	− 0	+ 38	+ 75
	− 78	− 48	− 24	+ 0	− 345	− 215	−145	−105	− 75	− 37	+ 0
6·01– 10	− 40	− 0	+ 29	+ 58	− 280	− 150	− 80	− 40	− 0	+ 45	+ 90
	− 98	− 58	− 29	+ 0	− 370	− 240	−170	−130	− 90	− 45	+ 0
10·01– 14	− 50	− 0	+ 35	+ 70	− 290	− 150	− 95	− 50	− 0	+ 55	+110
	−120	− 70	− 35	+ 0	− 400	− 260	−205	−160	−110	− 55	+ 0
14·01– 18	− 50	− 0	+ 35	+ 70	− 290	− 150	− 95	− 50	− 0	+ 55	+110
	−120	− 70	− 35	+ 0	− 400	− 260	−205	−160	−110	− 55	+ 0
18·01– 24	− 65	− 0	+ 42	+ 84	− 300	− 160	−110	− 65	− 0	+ 65	+130
	−149	− 84	− 42	+ 0	− 430	− 290	−240	−195	−130	− 65	+ 0
24·01– 30	− 65	− 0	+ 42	+ 84	− 300	− 160	−110	− 65	− 0	+ 65	+130
	−149	− 84	− 42	+ 0	− 430	− 290	−240	−195	−130	− 65	+ 0
30·01– 40	− 80	− 0	+ 50	+100	− 310	− 170	−120	− 80	− 0	+ 80	+160
	−180	−100	− 50	+ 0	− 470	− 330	−280	−240	−160	− 80	+ 0
40·01– 50	− 80	− 0	+ 50	+100	− 320	− 180	−130	− 80	− 0	+ 80	+160
	−180	−100	− 50	+ 0	− 480	− 340	−290	−240	−160	− 80	+ 0
50·01– 65	−100	− 0	+ 60	+120	− 340	− 190	−140	−100	− 0	+ 95	+190
	−220	−120	− 60	+ 0	− 530	− 380	−330	−290	−190	− 95	+ 0
65·01– 80	−100	− 0	+ 60	+120	− 360	− 200	−150	−100	− 0	+ 95	+190
	−220	−120	− 60	+ 0	− 550	− 390	−340	−290	−190	− 95	+ 0
80·01–100	−120	− 0	+ 70	+140	− 380	− 220	−170	−120	− 0	+110	+220
	−260	−140	− 70	+ 0	− 600	− 440	−390	−340	−220	−110	+ 0
100·1 –120	−120	− 0	+ 70	+140	− 410	− 240	−180	−120	− 0	+110	+220
	−260	−140	− 70	+ 0	− 630	− 460	−400	−340	−220	−110	+ 0
120·1 –140	−145	− 0	+ 80	+160	− 460	− 260	−200	−145	− 0	+125	+250
	−305	−160	− 80	+ 0	− 710	− 510	−450	−395	−250	−125	+ 0
140·1 –160	−145	− 0	+ 80	+160	− 520	− 280	−210	−145	− 0	+125	+250
	−305	−160	− 80	+ 0	− 770	− 530	−460	−395	−250	−125	+ 0
160·1 –180	−145	− 0	+ 80	+160	− 580	− 310	−230	−145	− 0	+125	+250
	−305	−160	− 80	+ 0	− 830	− 560	−480	−395	−250	−125	+ 0
180·1 –200	−170	− 0	+ 93	+185	− 660	− 340	−240	−170	− 0	+145	+290
	−355	−185	− 92	+ 0	− 950	− 630	−530	−460	−290	−145	+ 0
200·1 –225	−170	− 0	+ 93	+185	− 740	− 380	−260	−170	− 0	+145	+290
	−355	−185	− 92	+ 0	−1030	− 670	−550	−460	−290	−145	+ 0
225·1 –250	−170	− 0	+ 93	+185	− 820	− 420	−280	−170	− 0	+145	+290
	−355	−185	− 92	+ 0	−1110	− 710	−570	−460	−290	−145	+ 0
250·1 –280	−190	− 0	+105	+210	− 920	− 480	−300	−190	− 0	+160	+320
	−400	−210	−105	+ 0	−1240	− 800	−620	−510	−320	−160	+ 0
280·1 –315	−190	− 0	+105	+210	−1050	− 540	−330	−190	− 0	+160	+320
	−400	−210	−105	+ 0	−1370	− 860	−650	−510	−320	−160	+ 0
315·1 –355	−210	− 0	+115	+230	−1200	− 600	−360	−210	− 0	+180	+360
	−440	−230	−115	+ 0	−1560	− 960	−720	−570	−360	−180	+ 0
355·1 –400	−210	− 0	+115	+230	−1350	− 680	−400	−210	− 0	+180	+360
	−440	−230	−115	+ 0	−1710	−1040	−760	−570	−360	−180	+ 0
400·1 –450	−230	− 0	+125	+250	−1500	− 760	−440	−230	− 0	+200	+400
	−480	−250	−125	+ 0	−1900	−1160	−840	−630	−400	−200	+ 0
450·1 –500	−230	− 0	+125	+250	−1650	− 840	−480	−230	− 0	+200	+400
	−480	−250	−125	+ 0	−2050	−1240	−880	−630	−400	−200	+ 0

Quality 6

TABLE 23

I.S.A. Limits—Holes: Inch System

Diameter—in.	F6	G6	H6	J6	K6	M6	N6	P6	R6	S6	T6	U6	(V6)	X6
0·041– 0·16	+ 0·55 + 0·3	+ 0·35 + 0·1	+ 0·25 + 0	+ 0·1 − 0·15	—	− 0·05 − 0·3	− 0·25 − 0·5	− 0·35 − 0·6	− 0·45 − 0·7	− 0·55 − 0·8	—	− 0·65 − 0·9	—	− 0·85 − 1·1
0·161– 0·315	+ 0·7 + 0·4	+ 0·5 + 0·2	+ 0·3 + 0	+ 0·15 − 0·15	—	− 0·1 − 0·4	− 0·3 − 0·6	− 0·4 − 0·7	− 0·6 − 0·9	− 0·8 − 1·1	—	− 0·8 − 1·1	—	− 1·0 − 1·3
0·316– 0·4	+ 1·0 + 0·6	+ 0·6 + 0·2	+ 0·4 + 0	+ 0·25 − 0·15	+ 0·15 − 0·25	− 0·1 − 0·45	− 0·35 − 0·75	− 0·5 − 0·9	− 0·7 − 1·1	− 0·9 − 1·3	—	− 1·0 − 1·4	—	− 1·2 − 1·6
0·401– 0·5	+ 1·0 + 0·6	+ 0·6 + 0·2	+ 0·4 + 0	+ 0·25 − 0·15	+ 0·15 − 0·25	− 0·1 − 0·45	− 0·35 − 0·75	− 0·5 − 0·9	− 0·7 − 1·1	− 0·9 − 1·3	—	− 1·2 − 1·6	—	− 1·4 − 1·8
0·501– 0·63	+ 1·15 + 0·7	+ 0·75 + 0·3	+ 0·45 + 0	+ 0·25 − 0·2	+ 0·15 − 0·3	− 0·1 − 0·55	− 0·45 − 0·9	− 0·55 − 1·0	− 0·75 − 1·2	− 1·05 − 1·5	—	− 1·35 − 1·8	− 1·35 − 1·8	− 1·55 − 2·0
0·631– 0·8	+ 1·15 + 0·7	+ 0·75 + 0·3	+ 0·45 + 0	+ 0·25 − 0·2	+ 0·15 − 0·3	− 0·1 − 0·55	− 0·45 − 0·9	− 0·55 − 1·0	− 0·75 − 1·2	− 1·05 − 1·5	—	− 1·35 − 1·8	− 1·55 − 2·0	− 1·85 − 2·3
0·801– 1·0	+ 1·3 + 0·8	+ 0·8 + 0·3	+ 0·5 + 0	+ 0·3 − 0·2	+ 0·15 − 0·35	− 0·2 − 0·65	− 0·5 − 1·0	− 0·8 − 1·3	− 1·0 − 1·5	− 1·3 − 1·8	—	− 1·6 − 2·1	− 1·9 − 2·4	− 2·2 − 2·7
1·001– 1·25	+ 1·3 + 0·8	+ 0·8 + 0·3	+ 0·5 + 0	+ 0·3 − 0·2	+ 0·15 − 0·35	− 0·2 − 0·65	− 0·5 − 1·0	− 0·8 − 1·3	− 1·0 − 1·5	− 1·3 − 1·8	− 1·6 − 2·1	− 1·9 − 2·4	− 2·1 − 2·6	− 2·5 − 3·0
1·251– 1·6	+ 1·6 + 1·0	+ 0·9 + 0·3	+ 0·6 + 0	+ 0·35 − 0·25	+ 0·15 − 0·5	− 0·2 − 0·85	− 0·6 − 1·2	− 1·0 − 1·6	− 1·3 − 1·9	− 1·7 − 2·3	− 1·9 − 2·5	− 2·3 − 2·9	− 2·7 − 3·3	− 3·2 − 3·8
1·601– 2·0	+ 1·6 + 1·0	+ 0·9 + 0·3	+ 0·6 + 0	+ 0·35 − 0·25	+ 0·15 − 0·5	− 0·2 − 0·85	− 0·6 − 1·2	− 1·0 − 1·6	− 1·3 − 1·9	− 1·7 − 2·3	− 2·1 − 2·7	− 2·7 − 3·3	− 3·1 − 3·7	− 3·8 − 4·4
2·001– 2·5	+ 1·9 + 1·2	+ 1·1 + 0·4	+ 0·7 + 0	+ 0·45 − 0·25	+ 0·15 − 0·6	− 0·2 − 1·0	− 0·6 − 1·3	− 1·1 − 1·8	− 1·4 − 2·1	− 1·9 − 2·6	− 2·4 − 3·1	− 3·2 − 3·9	− 3·8 − 4·5	− 4·6 − 5·3
2·501– 3·15	+ 1·9 + 1·2	+ 1·1 + 0·4	+ 0·7 + 0	+ 0·45 − 0·25	+ 0·15 − 0·6	− 0·2 − 1·0	− 0·6 − 1·3	− 1·1 − 1·8	− 1·5 − 2·2	− 2·1 − 2·8	− 2·7 − 3·4	− 3·8 − 4·5	− 4·5 − 5·2	− 5·3 − 6·0
3·151– 4·0	+ 2·3 + 1·4	+ 1·4 + 0·5	+ 0·9 + 0	+ 0·6 − 0·25	+ 0·2 − 0·7	− 0·3 − 1·1	− 0·7 − 1·6	− 1·2 − 2·1	− 1·7 − 2·6	− 2·5 − 3·4	− 3·3 − 4·2	− 4·6 − 5·5	− 5·7 − 6·6	− 6·7 − 7·6
4·001– 5·0	+ 2·3 + 1·4	+ 1·4 + 0·5	+ 0·9 + 0	+ 0·6 − 0·25	+ 0·2 − 0·7	− 0·3 − 1·1	− 0·7 − 1·6	− 1·2 − 2·1	− 1·9 − 2·8	− 2·9 − 3·8	− 3·9 − 4·8	− 5·7 − 6·6	− 6·7 − 7·6	− 8·2 − 9·1
5·001– 5·6	+ 2·7 + 1·7	+ 1·5 + 0·5	+ 1·0 + 0	+ 0·8 − 0·25	+ 0·2 − 0·8	− 0·3 − 1·3	− 0·8 − 1·8	− 1·4 − 2·4	− 2·2 − 3·2	− 3·4 − 4·4	− 4·6 − 5·6	− 6·7 − 7·7	− 7·7 − 8·7	− 9·7 − 10·7
5·601– 6·3	+ 2·7 + 1·7	+ 1·5 + 0·5	+ 1·0 + 0	+ 0·8 − 0·25	+ 0·2 − 0·8	− 0·3 − 1·3	− 0·8 − 1·8	− 1·4 − 2·4	− 2·3 − 3·3	− 3·6 − 4·6	− 5·2 − 6·2	− 7·2 − 8·2	− 8·7 − 9·7	− 10·7 − 11·7
6·301– 7·1	+ 2·9 + 1·8	+ 1·7 + 0·6	+ 1·1 + 0	+ 0·9 − 0·25	+ 0·2 − 0·9	− 0·3 − 1·4	− 0·9 − 2·0	− 1·5 − 2·6	− 2·5 − 3·6	− 4·1 − 5·2	− 5·7 − 6·8	− 8·2 − 9·3	− 9·7 − 10·8	− 11·7 − 12·8
7·101– 8·0	+ 2·9 + 1·8	+ 1·7 + 0·6	+ 1·1 + 0	+ 0·9 − 0·25	+ 0·2 − 0·9	− 0·3 − 1·4	− 0·9 − 2·0	− 1·5 − 2·6	− 2·6 − 3·7	− 4·4 − 5·5	− 6·2 − 7·3	− 9·2 − 10·3	− 10·7 − 11·8	− 13·7 − 14·8
8·001– 9·0	+ 3·2 + 2·0	+ 1·8 + 0·6	+ 1·2 + 0	+ 1·0 − 0·25	+ 0·2 − 1·0	− 0·3 − 1·5	− 0·9 − 2·1	− 1·6 − 2·8	− 2·9 − 4·1	− 5·1 − 6·3	− 6·6 − 7·8	− 9·6 − 10·8	− 11·6 − 12·8	− 14·6 − 15·8
9·001–10·0	+ 3·2 + 2·0	+ 1·8 + 0·6	+ 1·2 + 0	+ 1·0 − 0·25	+ 0·2 − 1·0	− 0·3 − 1·5	− 0·9 − 2·1	− 1·6 − 2·8	− 3·0 − 4·2	− 5·1 − 6·3	− 7·6 − 8·8	− 10·6 − 11·8	− 13·6 − 14·8	− 16·6 − 17·8
10·001–11·2	+ 3·5 + 2·2	+ 2·0 + 0·7	+ 1·3 + 0	+ 1·1 − 0·25	+ 0·2 − 1·1	− 0·3 − 1·6	− 1·0 − 2·3	− 1·8 − 3·1	− 3·3 − 4·6	− 6·1 − 7·4	− 8·1 − 9·4	− 12·6 − 13·9	− 14·6 − 15·9	− 18·6 − 19·9
11·201–12·5	+ 3·5 + 2·2	+ 2·0 + 0·7	+ 1·3 + 0	+ 1·1 − 0·25	+ 0·2 − 1·1	− 0·3 − 1·6	− 1·0 − 2·3	− 1·8 − 3·1	− 3·4 − 4·7	− 6·6 − 7·9	− 9·1 − 10·4	− 13·6 − 14·9	− 16·6 − 17·9	− 20·6 − 21·9
12·501–14·0	+ 3·8 + 2·4	+ 2·1 + 0·7	+ 1·4 + 0	+ 1·1 − 0·3	+ 0·2 − 1·2	− 0·4 − 1·8	− 1·1 − 2·5	− 2·0 − 3·4	− 3·9 − 5·3	− 7·1 − 8·5	− 10·6 − 12·0	− 14·6 − 16·0	− 18·6 − 20·0	− 22·6 − 24·0
14·001–16·0	+ 3·8 + 2·4	+ 2·1 + 0·7	+ 1·4 + 0	+ 1·1 − 0·3	+ 0·2 − 1·2	− 0·4 − 1·8	− 1·1 − 2·5	− 2·0 3·4	− 4·1 − 5·5	− 7·6 − 9·0	− 11·6 − 13·0	− 16·6 − 18·0	− 20·6 − 22·0	− 25·6 − 27·0
16·001–18·0	+ 4·2 + 2·7	+ 2·3 + 0·8	+ 1·5 + 0	+ 1·2 − 0·3	+ 0·2 − 1·3	− 0·4 − 1·9	− 1·2 − 2·7	− 2·3 − 3·8	− 4·6 − 6·1	− 9·1 − 10·6	− 12·6 − 14·1	− 18·6 − 20·1	− 23·6 − 25·1	− 29·5 − 31·0
18·001–20·0	+ 4·2 + 2·7	+ 2·3 + 0·8	+ 1·5 + 0	+ 1·2 − 0·3	+ 0·2 − 1·3	− 0·4 − 1·9	− 1·2 − 2·7	− 2·3 − 3·8	− 4·6 − 6·1	− 9·6 − 11·1	− 13·6 − 15·1	− 20·6 − 22·1	− 25·6 − 27·1	− 32·5 − 34·0

Tolerance unit = 0·001 in.

TABLE 23 (*contd.*)
I.S.A. Limits—Holes: Inch System

Quality 7

Diameter—in.	E7	F7	G7	H7	J7	K7	M7	N7	P7	R7	S7	T7	U7	(V7)	X7	(Y7)	Z7
0·041– 0·16	+0·95 +0·6	+0·65 +0·3	+0·45 +0·1	+0·35 +0	+0·15 −0·2	—	−0 −0·35	−0·2 −0·55	−0·3 −0·65	−0·4 −0·75	−0·5 −0·85	—	−0·6 −0·95	—	−0·8 −1·15	—	−1·1 −1·45
0·161– 0·315	+1·4 +0·9	+0·9 +0·4	+0·7 +0·2	+0·5 +0	+0·25 −0·25	—	−0 −0·5	−0·2 −0·7	−0·3 −0·8	−0·5 −1·0	−0·7 −1·2	—	−0·7 −1·2	—	−0·9 −1·4	—	−1·3 −1·8
0·316– 0·4	+1·7 +1·1	+1·2 +0·6	+0·8 +0·2	+0·6 +0	+0·3 −0·3	+0·2 −0·4	−0 −0·6	−0·3 −0·9	−0·4 −1·0	−0·6 −1·2	−0·8 −1·4	—	−0·9 −1·5	—	−1·1 −1·7	—	−1·6 −2·2
0·401– 0·5	+1·7 +1·1	+1·2 +0·6	+0·8 +0·2	+0·6 +0	+0·3 −0·3	+0·2 −0·4	−0 −0·6	−0·3 −0·9	−0·4 −1·0	−0·6 −1·2	−0·8 −1·4	—	−1·1 −1·7	—	−1·3 −1·9	—	−1·8 −2·4
0·501– 0·63	+2·15 +1·4	+1·4 +0·7	+1·0 +0·3	+0·7 +0	+0·35 −0·35	+0·25 −0·45	−0 −0·7	−0·3 −1·0	−0·5 −1·2	−0·7 −1·4	−1·0 −1·7	—	−1·3 −2·0	−1·3 −2·0	−1·5 −2·2	—	−2·0 −2·7
0·631– 0·8	+2·15 +1·4	+1·4 +0·7	+1·0 +0·3	+0·7 +0	+0·35 −0·35	+0·25 −0·45	−0 −0·7	−0·3 −1·0	−0·5 −1·2	−0·7 −1·4	−1·0 −1·7	—	−1·3 −2·0	−1·5 −2·2	−1·8 −2·5	−2·0 −2·7	−2·4 −3·1
0·801– 1·0	+2·45 +1·6	+1·6 +0·8	+1·1 +0·3	+0·8 +0	+0·45 −0·35	+0·3 −0·5	−0 −0·8	−0·3 −1·1	−0·6 −1·4	−0·8 −1·6	−1·1 −1·9	—	−1·4 −2·2	−1·7 −2·5	−2·0 −2·8	−2·3 −3·1	−2·8 −3·6
1·001– 1·25	+2·45 +1·6	+1·6 +0·8	+1·1 +0·3	+0·8 +0	+0·45 −0·35	+0·3 −0·5	−0 −0·8	−0·3 −1·1	−0·6 −1·4	−0·8 −1·6	−1·1 −1·9	−1·4 −2·2	−1·7 −2·5	−1·9 −2·7	−2·3 −3·1	−2·8 −3·6	−3·3 −4·1
1·251– 1·6	+3·0 +2·0	+2·0 +1·0	+1·3 +0·3	+1·0 +0	+0·6 −0·4	+0·3 −0·7	−0 −1·0	−0·3 −1·3	−0·7 −1·7	−1·0 −2·0	−1·4 −2·4	−1·6 −2·6	−2·0 −3·0	−2·4 −3·4	−2·9 −3·9	−3·0 −4·0	−4·1 −5·1
1·601– 2·0	+3·0 +2·0	+2·0 +1·0	+1·3 +0·3	+1·0 +0	+0·6 −0·4	+0·3 −0·7	−0 −1·0	−0·3 −1·3	−0·7 −1·7	−1·0 −2·0	−1·4 −2·4	−1·8 −2·8	−2·4 −3·4	−2·8 −3·8	−3·5 −4·5	−4·2 −5·2	−5·1 −6·1
2·001– 2·5	+3·6 +2·4	+2·4 +1·2	+1·6 +0·4	+1·2 +0	+0·75 −0·45	+0·4 −0·8	−0 −1·2	−0·4 −1·5	−0·8 −2·0	−1·1 −2·3	−1·6 −2·8	−2·1 −3·3	−2·9 −4·1	−3·5 −4·7	−4·3 −5·5	−5·0 −6·2	−6·5 −7·7
2·501– 3·15	+3·6 +2·4	+2·4 +1·2	+1·6 +0·4	+1·2 +0	+0·75 −0·45	+0·4 −0·8	−0 −1·2	−0·4 −1·5	−0·8 −2·0	−1·2 −2·4	−1·8 −3·0	−2·4 −3·6	−3·5 −4·7	−4·2 −5·4	−5·0 −6·2	−6·5 −7·7	−7·5 −8·7
3·151– 4·0	+4·3 +2·9	+2·8 +1·4	+1·9 +0·5	+1·4 +0	+0·9 −0·5	+0·4 −1·0	−0 −1·4	−0·4 −1·9	−1·0 −2·4	−1·5 −2·9	−2·3 −3·7	−3·1 −4·5	−4·4 −5·8	−5·5 −6·9	−6·5 −7·9	−8·0 −9·4	−9·5 −10·9
4·001– 5·0	+4·3 +2·9	+2·8 +1·4	+1·9 +0·5	+1·4 +0	+0·9 −0·5	+0·4 −1·0	−0 −1·4	−0·4 −1·9	−1·0 −2·4	−1·7 −3·1	−2·7 −4·1	−3·7 −5·1	−5·5 −6·9	−6·5 −7·9	−7·9 −9·4	−9·5 −10·9	−12·5 −13·9
5·001– 5·6	+4·9 +3·3	+3·3 +1·7	+2·1 +0·5	+1·6 +0	+1·0 −0·6	+0·5 −1·1	−0 −1·6	−0·5 −2·1	−1·1 −2·7	−1·9 −3·5	−3·1 −4·7	−4·3 −5·9	−6·4 −8·0	−7·4 −9·0	−9·4 −11·0	−11·4 −13·0	−14·4 −16·0
5·601– 6·3	+4·9 +3·3	+3·3 +1·7	+2·1 +0·5	+1·6 +0	+1·0 −0·6	+0·5 −1·1	−0 −1·6	−0·5 −2·1	−1·1 −2·7	−2·0 −3·6	−3·3 −4·9	−4·9 −6·5	−6·9 −8·5	−8·4 −10·0	−10·4 −12·0	−12·4 −14·0	−15·4 −17·0
6·301– 7·1	+5·3 +3·6	+3·5 +1·8	+2·3 +0·6	+1·7 +0	+1·1 −0·6	+0·5 −1·2	−0 −1·7	−0·6 −2·3	−1·2 −2·9	−2·2 −3·9	−3·8 −5·5	−5·4 −7·1	−7·9 −9·6	−9·4 −11·1	−11·4 −13·1	−14·4 −16·1	−17·4 −19·1
7·101– 8·0	+5·3 +3·6	+3·5 +1·8	+2·3 +0·6	+1·7 +0	+1·1 −0·6	+0·5 −1·2	−0 −1·7	−0·6 −2·3	−1·2 −2·9	−2·3 −4·0	−4·1 −5·8	−5·9 −7·6	−8·9 −10·6	−10·4 −12·1	−13·4 −15·1	−16·4 −18·1	−20·4 −22·1
8·001– 9·0	+5·9 +4·0	+3·9 +2·0	+2·5 +0·6	+1·9 +0	+1·3 −0·6	+0·6 −1·3	−0 −1·9	−0·6 −2·5	−1·3 −3·2	−2·6 −4·5	−4·8 −6·7	−6·3 −8·2	−9·3 −11·2	−11·3 −13·2	−14·3 −16·2	−18·3 −20·2	−22·3 −24·2
9·001–10·0	+5·9 +4·0	+3·9 +2·0	+2·5 +0·6	+1·9 +0	+1·3 −0·6	+0·6 −1·3	−0 −1·9	−0·6 −2·5	−1·3 −3·2	−2·7 −4·6	−4·8 −6·7	−7·3 −9·2	−10·3 −12·2	−13·3 −15·2	−16·3 −18·2	−20·3 −22·2	−25·3 −27·2
10·001–11·2	+6·4 +4·4	+4·2 +2·2	+2·7 +0·7	+2·0 +0	+1·4 −0·6	+0·6 −1·4	−0 −2·0	−0·7 −2·7	−1·5 −3·5	−3·0 −5·0	−5·8 −7·8	−7·8 −9·8	−12·3 −14·3	−14·3 −16·3	−18·3 −20·3	−22·3 −24·3	−27·3 −29·3
11·201–12·5	+6·4 +4·4	+4·2 +2·2	+2·7 +0·7	+2·0 +0	+1·4 −0·6	+0·6 −1·4	−0 −2·0	−0·7 −2·7	−1·5 −3·5	−3·1 −5·1	−6·3 −8·3	−8·8 −10·8	−13·3 −15·3	−16·3 −18·3	−20·3 −22·3	−25·3 −27·3	−31·3 −33·3
12·501–14·0	+7·0 +4·8	+4·6 +2·4	+2·9 +0·7	+2·2 +0	+1·5 −0·7	+0·7 −1·5	−0 −2·2	−0·7 −2·9	−1·6 −3·8	−3·5 −5·7	−6·7 −8·9	−10·2 −12·4	−14·2 −16·4	−18·2 −20·4	−22·2 −24·4	−28·2 −30·4	−34·2 −36·4
14·001–16·0	+7·0 +4·8	+4·6 +2·4	+2·9 +0·7	+2·2 +0	+1·5 −0·7	+0·7 −1·5	−0 −2·2	−0·7 −2·9	−1·6 −3·8	−3·7 −5·9	−7·2 −9·4	−11·2 −13·4	−16·2 −18·4	−20·2 −22·4	−25·2 −27·4	−31·2 −33·4	−39·2 −41·4
16·001–18·0	+8·0 +5·5	+5·2 +2·7	+3·3 +0·8	+2·5 +0	+1·8 −0·7	+0·8 −1·7	−0 −2·5	−0·7 −3·1	−1·7 −4·2	−4·0 −6·5	−8·5 −11·0	−12·0 −14·5	−18·0 −20·5	−23·0 −25·5	−29·0 −31·5	−35·0 −37·5	−44·0 −46·5
18·001–20·0	+8·0 +5·5	+5·2 +2·7	+3·3 +0·8	+2·5 +0	+1·8 −0·7	+0·8 −1·7	−0 −2·5	−0·7 −3·1	−1·7 −4·2	−4·0 −6·5	−9·0 −11·5	−13·0 −15·5	−20·0 −22·5	−25·0 −27·5	−32·0 −34·5	−39·0 −41·5	−49·0 −51·5

Tolerance unit = 0·001 in

TABLE 23 (*contd.*)

I.S.A. LIMITS—HOLES: INCH SYSTEM

Quality 8

Diameter—in.	B8	C8	D8	E8	F8	H8	J8	K8	M8	N8
0·041– 0·16	+ 6·1	+ 3·0	+ 1·5	+ 1·2	+ 0·9	+ 0·6	+ 0·35	—	—	− 0·05
	+ 5·5	+ 2·4	+ 0·9	+ 0·6	+ 0·3	+ 0	− 0·25			− 0·65
0·161– 0·315	+ 6·3	+ 3·7	+ 2·2	+ 1·7	+ 1·2	+ 0·8	+ 0·45	—	—	− 0·1
	+ 5·5	+ 2·0	+ 1·4	+ 0·9	+ 0·4	+ 0	− 0·35			− 0·9
0·316– 0·4	+ 7·0	+ 4·2	+ 2·7	+ 2·0	+ 1·5	+ 1·0	+ 0·6	+ 0·4	+ 0·1	− 0·1
	+ 6·0	+ 3·2	+ 1·7	+ 1·1	+ 0·6	+ 0	− 0·4	− 0·6	− 0·8	− 1·1
0·401– 0·5	+ 7·0	+ 4·2	+ 2·7	+ 2·1	+ 1·5	+ 1·0	+ 0·6	+ 0·4	+ 0·1	− 0·1
	+ 6·0	+ 3·2	+ 1·7	+ 1·1	+ 0·6	+ 0	− 0·4	− 0·6	− 0·8	− 1·1
0·501– 0·63	+ 7·1	+ 4·7	+ 3·2	+ 2·5	+ 1·8	+ 1·1	+ 0·7	+ 0·4	+ 0·1	− 0·1
	+ 6·0	+ 3·6	+ 2·1	+ 1·4	+ 0·7	+ 0	− 0·4	− 0·7	− 0·95	− 1·3
0·631– 0·8	+ 7·1	+ 4·7	+ 3·2	+ 2·5	+ 1·8	+ 1·1	+ 0·7	+ 0·4	+ 0·1	− 0·1
	+ 6·0	+ 3·6	+ 2·1	+ 1·4	+ 0·7	+ 0	− 0·4	− 0·7	− 0·95	− 1·3
0·801– 1·0	+ 7·8	+ 5·2	+ 3·9	+ 2·6	+ 2·1	+ 1·3	+ 0·8	+ 0·5	+ 0·2	− 0·1
	+ 6·5	+ 3·9	+ 2·6	+ 1·6	+ 0·8	+ 0	− 0·5	− 0·8	− 1·1	− 1·4
1·001– 1·25	+ 7·8	+ 5·2	+ 3·9	+ 2·6	+ 2·1	+ 1·3	+ 0·8	+ 0·5	+ 0·2	− 0·1
	+ 6·5	+ 3·9	+ 2·6	+ 1·6	+ 0·8	+ 0	− 0·5	− 0·8	− 1·1	− 1·4
1·251– 1·6	+ 8·1	+ 5·8	+ 4·8	+ 3·6	+ 2·6	+ 1·6	+ 1·0	+ 0·5	+ 0·2	− 0·1
	+ 6·5	+ 4·2	+ 3·2	+ 2·0	+ 1·0	+ 0	− 0·6	− 1·1	− 1·4	− 1·7
1·601– 2·0	+ 8·6	+ 6·0	+ 4·8	+ 3·6	+ 2·6	+ 1·6	+ 1·0	+ 0·5	+ 0·2	− 0·1
	+ 7·0	+ 4·4	+ 3·2	+ 2·0	+ 1·0	+ 0	− 0·6	− 1·1	− 1·4	− 1·7
2·001– 2·5	+ 9·3	+ 7·3	+ 5·7	+ 4·2	+ 3·0	+ 1·8	+ 1·1	+ 0·5	+ 0·2	− 0·1
	+ 7·5	+ 5·5	+ 3·9	+ 2·4	+ 1·2	+ 0	− 0·7	− 1·3	− 1·7	− 2·0
2·501– 3·15	+ 9·8	+ 7·8	+ 5·7	+ 4·2	+ 3·0	+ 1·8	+ 1·1	+ 0·5	+ 0·2	− 0·1
	+ 8·0	+ 6·0	+ 3·9	+ 2·4	+ 1·2	+ 0	− 0·7	− 1·3	− 1·7	− 2·0
3·151– 4·0	+ 10·7	+ 8·7	+ 7·0	+ 5·1	+ 3·6	+ 2·2	+ 1·4	+ 0·7	+ 0·3	− 0·2
	+ 8·5	+ 6·5	+ 4·8	+ 2·9	+ 1·4	+ 0	− 0·8	− 1·5	− 1·9	− 2·4
4·001– 5·0	+ 11·7	+ 9·7	+ 7·0	+ 5·1	+ 3·6	+ 2·2	+ 1·4	+ 0·7	+ 0·3	− 0·2
	+ 9·5	+ 7·5	+ 4·8	+ 2·9	+ 1·4	+ 0	− 0·8	− 1·5	− 1·9	− 2·4
5·001– 5·6	+ 12·9	+ 10·5	+ 8·0	+ 5·8	+ 4·2	+ 2·5	+ 1·6	+ 0·8	+ 0·3	− 0·2
	+ 9·5	+ 8·0	+ 5·5	+ 3·3	+ 1·7	+ 0	− 0·9	− 1·7	− 2·2	− 2·7
5·601– 6·3	+ 13·5	+ 11·0	+ 8·0	+ 5·8	+ 4·2	+ 2·5	+ 1·6	+ 0·8	+ 0·3	− 0·2
	+ 11·0	+ 8·5	+ 5·5	+ 3·3	+ 1·7	+ 0	− 0·9	− 1·7	− 2·2	− 2·7
6·301– 7·1	+ 14·7	+ 11·7	+ 8·7	+ 6·3	+ 4·5	+ 2·7	+ 1·8	+ 0·9	+ 0·3	− 0·2
	+ 12·0	+ 9·0	+ 6·0	+ 3·6	+ 1·8	+ 0	− 0·9	− 1·8	− 2·3	− 2·9
7·101– 8·0	+ 16·7	+ 12·7	+ 8·7	+ 6·3	+ 4·5	+ 2·7	+ 1·8	+ 0·9	+ 0·3	− 0·2
	+ 14·0	+ 10·0	+ 6·0	+ 3·6	+ 1·8	+ 0	− 0·9	− 1·8	− 2·3	− 2·9
8·001– 9·0	+ 17·9	+ 13·9	+ 9·9	+ 6·9	+ 4·9	+ 2·9	+ 1·9	+ 0·9	+ 0·4	− 0·2
	+ 15·0	+ 11·0	+ 7·0	+ 4·0	+ 2·0	+ 0	− 1·0	− 2·0	− 2·6	− 3·2
9·001–10·0	+ 19·9	+ 13·9	+ 9·9	+ 6·9	+ 4·9	+ 2·9	+ 1·9	+ 0·9	+ 0·4	− 0·2
	+ 17·0	+ 11·0	+ 7·0	+ 4·0	+ 2·0	+ 0	− 1·0	− 2·0	− 2·6	− 3·2
10·001–11·2	+ 22·2	+ 15·2	+ 10·7	+ 7·6	+ 5·4	+ 3·2	+ 2·2	+ 1·0	+ 0·4	− 0·2
	+ 19·0	+ 12·0	+ 7·5	+ 4·4	+ 2·2	+ 0	− 1·0	− 2·2	− 2·7	− 3·4
11·201–12·5	+ 24·2	+ 16·2	+ 10·7	+ 7·6	+ 5·4	+ 3·2	+ 2·2	+ 1·0	+ 0·4	− 0·2
	+ 21·0	+ 13·0	+ 7·5	+ 4·4	+ 2·2	+ 0	− 1·0	− 2·2	− 2·7	− 3·4
12·501–14·0	+ 27·5	+ 17·5	+ 12·0	+ 8·3	+ 5·9	+ 3·5	+ 2·4	+ 1·1	+ 0·5	− 0·2
	+ 24·0	+ 14·0	+ 8·5	+ 4·8	+ 2·4	+ 0	− 1·1	− 2·4	− 3·0	− 3·7
14·001–16·0	+ 30·5	+ 19·5	+ 12·0	+ 8·3	+ 5·9	+ 3·5	+ 2·4	+ 1·1	+ 0·5	− 0·2
	+ 27·0	+ 16·0	+ 8·5	+ 4·8	+ 2·4	+ 0	− 1·1	− 2·4	− 3·0	− 3·7
16·001–18·0	+ 34·9	+ 20·9	+ 13·4	+ 9·4	+ 6·6	+ 3·9	+ 2·6	+ 1·2	+ 0·6	− 0·2
	+ 31·0	+ 17·0	+ 9·5	+ 5·5	+ 2·7	+ 0	− 1·3	− 2·7	− 3·3	− 4·1
18·001–20·0	+ 37·9	+ 22·9	+ 13·4	+ 9·4	+ 6·6	+ 3·9	+ 2·6	+ 1·2	+ 0·6	− 0·2
	+ 34·0	+ 19·0	+ 9·5	+ 5·5	+ 2·7	+ 0	− 1·3	− 2·7	− 3·3	− 4·1

Tolerance unit = 0·001 in.

TABLE 23 (*contd.*)

I.S.A. LIMITS—HOLES: INCH SYSTEM

Quality 9

Diameter—in.	A9	B9	C9	D9	E9	F9	H9	J9	N9
0·041– 0·16	+ 11·9	+ 6·4	+ 3·3	+ 1·8	+ 1·5	+ 1·2	+ 0·9	+ 0·4	− 0
	+ 11·0	+ 5·5	+ 2·4	+ 0·9	+ 0·6	+ 0·3	+ 0	− 0·5	− 0·9
0·161– 0·315	+ 12·3	+ 6·8	+ 4·2	+ 2·7	+ 2·2	+ 1·7	+ 1·3	+ 0·6	− 0
	+ 11·0	+ 5·5	+ 2·9	+ 1·4	+ 0·9	+ 0·4	+ 0	− 0·7	− 1·3
0·316– 0·4	+ 12·5	+ 7·5	+ 4·7	+ 3·2	+ 2·6	+ 2·1	+ 1·5	+ 0·7	− 0
	+ 11·0	+ 6·0	+ 3·2	+ 1·7	+ 1·1	+ 0·6	+ 0	− 0·8	− 1·5
0·401– 0·5	+ 12·5	+ 7·5	+ 4·7	+ 3·2	+ 2·6	+ 2·1	+ 1·5	+ 0·7	− 0
	+ 11·0	+ 6·0	+ 3·2	+ 1·7	+ 1·1	+ 0·6	+ 0	− 0·8	− 1·5
0·501– 0·63	+ 12·8	+ 7·8	+ 5·4	+ 3·9	+ 3·2	+ 2·5	+ 1·8	+ 0·9	− 0
	+ 11·0	+ 6·0	+ 3·6	+ 2·1	+ 1·4	+ 0·7	+ 0	− 0·9	− 1·8
0·631– 0·8	+ 12·8	+ 7·8	+ 5·4	+ 3·9	+ 3·2	+ 2·5	+ 1·8	+ 0·9	− 0
	+ 11·0	+ 6·0	+ 3·6	+ 2·1	+ 1·4	+ 0·7	+ 0	− 0·9	− 1·8
0·801– 1·0	+ 14·1	+ 8·6	+ 6·0	+ 4·7	+ 3·7	+ 2·9	+ 2·1	+ 1·0	− 0
	+ 12·0	+ 6·5	+ 3·9	+ 2·6	+ 1·6	+ 0·8	+ 0	− 1·1	− 2·1
1·001– 1·25	+ 14·1	+ 8·6	+ 6·0	+ 4·7	+ 3·7	+ 2·9	+ 2·1	+ 1·0	− 0
	+ 12·0	+ 6·5	+ 3·9	+ 2·6	+ 1·6	+ 0·8	+ 0	− 1·1	− 2·1
1·251– 1·6	+ 14·5	+ 9·0	+ 6·7	+ 5·7	+ 4·5	—	+ 2·5	+ 1·2	− 0
	+ 12·0	+ 6·5	+ 4·2	+ 3·2	+ 2·0		+ 0	− 1·3	− 2·5
1·601– 2·0	+ 15·5	+ 9·5	+ 6·9	+ 5·7	+ 4·5	—	+ 2·5	+ 1·2	− 0
	+ 13·0	+ 7·0	+ 4·4	+ 3·2	+ 2·0		+ 0	− 1·3	− 2·5
2·001– 2·5	+ 15·9	+ 10·4	+ 8·4	+ 6·8	+ 5·3	—	+ 2·9	+ 1·4	− 0
	+ 13·0	+ 7·5	+ 5·5	+ 3·9	+ 2·4		+ 0	− 1·5	− 2·9
2·501– 3·15	+ 16·9	+ 10·9	+ 8·9	+ 6·8	+ 5·3	—	+ 2·9	+ 1·4	− 0
	+ 14·0	+ 8·0	+ 6·0	+ 3·9	+ 2·4		+ 0	− 1·5	− 2·9
3·151– 4·0	+ 18·5	+ 12·0	+ 10·0	+ 8·3	+ 6·4	—	+ 3·5	+ 1·7	− 0
	+ 15·0	+ 8·5	+ 6·5	+ 4·8	+ 2·9		+ 0	− 1·8	− 3·5
4·001– 5·0	+ 19·5	+ 13·0	+ 11·0	+ 8·3	+ 6·4	—	+ 3·5	+ 1·7	− 0
	+ 16·0	+ 9·5	+ 7·5	+ 4·8	+ 2·9		+ 0	− 1·8	− 3·5
5·001– 5·6	+ 22·9	+ 13·4	+ 11·9	+ 9·4	+ 7·2	—	+ 3·9	+ 1·9	− 0
	+ 19·0	+ 9·5	+ 8·0	+ 5·5	+ 3·3		+ 0	− 2·0	− 3·9
5·601– 6·3	+ 24·9	+ 14·9	+ 12·4	+ 9·4	+ 7·2	—	+ 3·9	+ 1·9	− 0
	+ 21·0	+ 11·0	+ 8·5	+ 5·5	+ 3·3		+ 0	− 2·0	− 3·9
6·301– 7·1	+ 27·3	+ 16·3	+ 13·3	+ 10·3	+ 7·9	—	+ 4·3	+ 2·1	− 0
	+ 23·0	+ 12·0	+ 9·0	+ 6·0	+ 3·6		+ 0	− 2·2	− 4·3
7·101– 8·0	+ 30·3	+ 18·3	+ 14·3	+ 10·3	+ 7·9	—	+ 4·3	+ 2·1	− 0
	+ 26·0	+ 14·0	+ 10·0	+ 6·0	+ 3·6		+ 0	− 2·2	− 4·3
8·001– 9·0	+ 34·7	+ 19·7	+ 15·7	+ 11·7	+ 8·7	—	+ 4·7	+ 2·3	− 0
	+ 30·0	+ 15·0	+ 11·0	+ 7·0	+ 4·0		+ 0	− 2·4	− 4·7
9·001–10·0	+ 37·7	+ 21·7	+ 15·7	+ 11·7	+ 8·7	—	+ 4·7	+ 2·3	− 0
	+ 33·0	+ 17·0	+ 11·0	+ 7·0	+ 4·0		+ 0	− 2·4	− 4·7
10·001–11·2	+ 42·0	+ 24·0	+ 17·0	+ 12·5	+ 9·4	—	+ 5·0	+ 2·5	− 0
	+ 37·0	+ 19·0	+ 12·0	+ 7·5	+ 4·4		+ 0	− 2·5	− 5·0
11·201–12·5	+ 46·0	+ 26·0	+ 18·0	+ 12·5	+ 9·4	—	+ 5·0	+ 2·5	− 0
	+ 41·0	+ 21·0	+ 13·0	+ 7·5	+ 4·4		+ 0	− 2·5	− 5·0
12·501–14·0	+ 51·5	+ 29·5	+ 19·5	+ 14·0	+ 10·3	—	+ 5·5	+ 2·7	− 0
	+ 46·0	+ 24·0	+ 14·0	+ 8·5	+ 4·8		+ 0	− 2·8	− 5·5
14·001–16·0	+ 57·5	+ 32·5	+ 21·5	+ 14·0	+ 10·3	—	+ 5·5	+ 2·7	− 0
	+ 52·0	+ 27·0	+ 16·0	+ 8·5	+ 4·8		+ 0	− 2·8	− 5·5
16·001–18·0	+ 65·0	+ 37·0	+ 23·0	+ 15·5	+ 11·5	—	+ 6·0	+ 3·0	− 0
	+ 59·0	+ 31·0	+ 17·0	+ 9·5	+ 5·5		+ 0	− 3·0	− 6·0
18·001–20·0	+ 72·0	+ 40·0	+ 25·0	+ 15·5	+ 11·5	—	+ 6·0	+ 3·0	− 0
	+ 66·0	+ 34·0	+ 19·0	+ 9·5	+ 5·5		+ 0	− 3·0	− 6·0

Tolerance unit = 0·001 in.

TABLE 23 (*contd.*)

I.S.A. Limits—Holes: Inch System

Qualities 10 and 11

Diameter—in.	D10	H10	J10	N10	A11	B11	C11	D11	H11	J11	N11
0·041– 0·16	+ 2·3 + 0·9	+ 1·4 + 0	+ 0·7 − 0·7	− 0 − 1·4	+ 13·3 + 11·0	+ 7·8 + 5·5	+ 4·7 + 2·4	+ 3·2 + 0·9	+ 2·3 + 0	+ 1·1 − 1·2	− 0 − 2·3
0·161– 0·315	+ 3·4 + 1·4	+ 2·0 + 0	+ 1·0 − 1·0	− 0 − 2·0	+ 14·2 + 11·0	+ 8·7 + 5·5	+ 6·1 + 2·9	+ 4·6 + 1·4	+ 3·2 + 0	+ 1·6 − 1·6	− 0 − 3·2
0·316– 0·4	+ 4·2 + 1·7	+ 2·5 + 0	+ 1·2 − 1·3	− 0 − 2·5	+ 14·9 + 11·0	+ 9·9 + 6·0	+ 7·1 + 3·2	+ 5·6 + 1·7	+ 3·9 + 0	+ 1·9 − 2·0	− 0 − 3·9
0·401– 0·5	+ 4·2 + 1·7	+ 2·5 + 0	+ 1·2 − 1·3	− 0 − 2·5	+ 14·9 + 11·0	+ 9·9 + 6·0	+ 7·1 + 3·2	+ 5·6 + 1·7	+ 3·9 + 0	+ 1·9 − 2·0	− 0 − 3·9
0·501– 0·63	+ 5·0 + 2·1	+ 2·9 + 0	+ 1·4 − 1·5	− 0 − 2·9	+ 15·5 + 11·0	+ 10·5 + 6·0	+ 8·1 + 3·6	+ 6·6 + 2·1	+ 4·5 + 0	+ 2·2 − 2·3	− 0 − 4·5
0·631 - 0·8	+ 5·0 + 2·1	+ 2·9 + 0	+ 1·4 − 1·5	− 0 − 2·9	+ 15·5 + 11·0	+ 10·5 + 6·0	+ 8·1 + 3·6	+ 6·6 + 2·1	+ 4·5 + 0	+ 2·2 − 2·3	− 0 − 4·5
0·801– 1·0	+ 6·0 + 2·6	+ 3·4 + 0	+ 1·7 − 1·7	− 0 − 3·4	+ 17·5 + 12·0	+ 12·0 + 6·5	+ 9·4 + 3·9	+ 8·1 + 2·6	+ 5·5 + 0	+ 2·7 − 2·8	− 0 − 5·5
1·001– 1·25	+ 6·0 + 2·6	+ 3·4 + 0	+ 1·7 − 1·7	− 0 − 3·4	+ 17·5 + 12·0	+ 12·0 + 6·5	+ 9·4 + 3·9	+ 8·1 + 2·6	+ 5·5 + 0	+ 2·7 − 2·8	− 0 − 5·5
1·251– 1·6	+ 7·2 + 3·2	+ 4·0 + 0	+ 2·0 − 2·0	− 0 − 4·0	+ 18·0 + 12·0	+ 12·5 + 6·5	+ 10·2 + 4·2	+ 9·2 + 3·2	+ 6·0 + 0	+ 3·0 − 3·0	− 0 − 6·0
1·601– 2·0	+ 7·2 + 3·2	+ 4·0 + 0	+ 2·0 − 2·0	− 0 − 4·0	+ 19·0 + 13·0	+ 13·0 + 7·0	+ 10·4 + 4·4	+ 9·2 + 3·2	+ 6·0 + 0	+ 3·0 − 3·0	− 0 − 6·0
2·001 - 2·5	+ 8·6 + 3·9	+ 4·7 + 0	+ 2·3 − 2·4	− 0 − 4·7	+ 20·5 + 13·0	+ 15·0 + 7·5	+ 13·0 + 5·5	+ 11·4 + 3·9	+ 7·5 + 0	+ 3·7 − 3·8	− 0 − 7·5
2·501– 3·15	+ 8·6 + 3·9	+ 4·7 + 0	+ 2·3 − 2·4	− 0 − 4·7	+ 21·5 + 14·0	+ 15·5 + 8·0	+ 13·5 + 6·0	+ 11·4 + 3·9	+ 7·5 + 0	+ 3·7 − 3·8	− 0 − 7·5
3·151– 4·0	+ 10·3 + 4·8	+ 5·5 + 0	+ 2·7 − 2·8	− 0 − 5·5	+ 23·5 + 15·0	+ 17·0 + 8·5	+ 15·0 + 6·5	+ 13·3 + 4·8	+ 8·5 + 0	+ 4·2 − 4·3	− 0 − 8·5
4·001– 5·0	+ 10·3 + 4·8	+ 5·5 + 0	+ 2·7 − 2·8	− 0 − 5·5	+ 24·5 + 16·0	+ 18·0 + 9·5	+ 16·0 + 7·5	+ 13·3 + 4·8	+ 8·5 + 0	+ 4·2 − 4·3	− 0 − 8·5
5·001 - 5·6	+ 12·0 + 5·5	+ 6·5 + 0	+ 3·2 − 3·3	− 0 − 6·5	+ 29·0 + 19·0	+ 19·5 + 9·5	+ 18·0 + 8·0	+ 15·5 + 5·5	+ 10·0 + 0	+ 5·0 − 5·0	− 0 − 10·0
5·601– 6·3	+ 12·0 + 5·5	+ 6·5 + 0	+ 3·2 − 3·3	− 0 − 6·5	+ 31·0 + 21·0	+ 21·0 + 11·0	+ 18·5 + 8·5	+ 15·5 + 5·5	+ 10·0 + 0	+ 5·0 − 5·0	− 0 − 10·0
6·301- 7·1	+ 13·0 + 6·0	+ 7·0 + 0	+ 3·5 − 3·5	− 0 − 7·0	+ 34·0 + 23·0	+ 23·0 + 12·0	+ 20·0 + 9·0	+ 17·0 + 6·0	+ 11·0 + 0	+ 5·5 − 5·5	− 0 − 11·0
7·101– 8·0	+ 13·0 + 6·0	+ 7·0 + 0	+ 3·5 − 3·5	− 0 − 7·0	+ 37·0 + 26·0	+ 25·0 + 14·0	+ 21·0 + 10·0	+ 17·0 + 6·0	+ 11·0 + 0	+ 5·5 − 5·5	− 0 − 11·0
8·001– 9·0	+ 14·5 + 7·0	+ 7·5 + 0	+ 3·7 − 3·8	− 0 − 7·5	+ 42·0 + 30·0	+ 27·0 + 15·0	+ 23·0 + 11·0	+ 19·0 + 7·0	+ 12·0 + 0	+ 6·0 − 6·0	− 0 − 12·0
9·001 -10·0	+ 14·5 + 7·0	+ 7·5 + 0	+ 3·7 − 3·8	− 0 − 7·5	+ 45·0 + 33·0	+ 29·0 + 17·0	+ 23·0 + 11·0	+ 19·0 + 7·0	+ 12·0 + 0	+ 6·0 − 6·0	− 0 − 12·0
10·001–11·2	+ 15·5 + 7·5	+ 8·0 + 0	+ 4·0 − 4·0	− 0 − 8·0	+ 50·0 + 37·0	+ 32·0 + 19·0	+ 25·0 + 12·0	+ 20·5 + 7·5	+ 13·0 + 0	+ 6·5 − 6·5	− 0 − 13·0
11·201–12·5	+ 15·5 + 7·5	+ 8·0 + 0	+ 4·0 − 4·0	− 0 − 8·0	+ 54·0 + 41·0	+ 34·0 + 21·0	+ 26·0 + 13·0	+ 20·5 + 7·5	+ 13·0 + 0	+ 6·5 − 6·5	− 0 − 13·0
12·501–14·0	+ 17·5 + 8·5	+ 9·0 + 0	+ 4·5 − 4·5	− 0 − 9·0	+ 60·0 + 46·0	+ 38·0 + 24·0	+ 28·0 + 14·0	+ 22·5 + 8·5	+ 14·0 + 0	+ 7·0 − 7·0	− 0 − 14·0
14·001–16·0	+ 17·5 + 8·5	+ 9·0 + 0	+ 4·5 − 4·5	− 0 − 9·0	+ 66·0 + 52·0	+ 41·0 + 27·0	+ 30·0 + 16·0	+ 22·5 + 8·5	+ 14·0 + 0	+ 7·0 − 7·0	− 0 − 14·0
16·001–18·0	+ 19·5 + 9·5	+ 10·0 + 0	+ 5·0 − 5·0	− 0 − 10·0	+ 74·0 + 59·0	+ 46·0 + 31·0	+ 32·0 + 17·0	+ 24·5 + 9·5	+ 15·0 + 0	+ 8·0 − 8·0	− 0 − 15·0
18·001 -20·0	+ 19·5 + 9·5	+ 10·0 + 0	+ 5·0 − 5·0	− 0 − 10·0	+ 81·0 + 66·0	+ 49·0 + 34·0	+ 34·0 + 19·0	+ 24·5 + 9·5	+ 15·0 + 0	+ 8·0 − 8·0	− 0 − 15·0

Tolerance unit = 0·001 in.

TABLE 24
I.S.A. LIMITS—SHAFTS: INCH SYSTEM

Quality 5

Diameter—in.	g5	h5	j5	k5	m5	n5	p5	r5	s5	t5	u5	(v5)	x5
0·041– 0·16	− 0·1 − 0·3	− 0 − 0·2	+ 0·15 − 0·05	—	+ 0·3 + 0·1	+ 0·5 + 0·3	+ 0·6 + 0·4	+ 0·7 + 0·5	+ 0·8 + 0·6	—	+ 0·9 + 0·7	—	+ 1·1 + 0·9
0·161– 0·315	− 0·2 − 0·4	− 0 − 0·2	+ 0·15 − 0·05	—	+ 0·4 + 0·2	+ 0·6 + 0·4	+ 0·7 + 0·5	+ 0·9 + 0·7	+ 1·1 + 0·9	—	+ 1·1 + 0·9	—	+ 1·3 + 1·1
0·316– 0·4	− 0·2 − 0·45	− 0 − 0·25	+ 0·15 − 0·1	+ 0·25 + 0	+ 0·45 + 0·2	+ 0·75 + 0·5	+ 0·85 + 0·6	+ 1·05 + 0·8	+ 1·25 + 1·1	—	+ 1·35 + 1·1	—	+ 1·55 + 1·3
0·401– 0·5	− 0·2 − 0·45	− 0 − 0·25	+ 0·15 − 0·1	+ 0·25 + 0	+ 0·45 + 0·2	+ 0·75 + 0·5	+ 0·85 + 0·6	+ 1·05 + 0·8	+ 1·25 + 1·1	—	+ 1·55 + 1·3	—	+ 1·75 + 1·5
0·501– 0·63	− 0·3 − 0·6	− 0 − 0·3	+ 0·2 − 0·1	+ 0·3 + 0	+ 0·55 + 0·25	+ 0·9 + 0·6	+ 1·0 + 0·7	+ 1·2 + 0·9	+ 1·5 + 1·2	—	+ 1·8 + 1·5	+ 1·8 + 1·5	+ 2·0 + 1·7
0·631– 0·8	− 0·3 − 0·6	− 0 − 0·3	+ 0·2 − 0·1	+ 0·3 + 0	+ 0·55 + 0·25	+ 0·9 + 0·6	+ 1·0 + 0·7	+ 1·2 + 0·9	+ 1·5 + 1·2	—	+ 1·8 + 1·5	+ 2·0 + 1·7	+ 2·3 + 2·0
0·801– 1·0	− 0·3 − 0·65	− 0 − 0·35	+ 0·2 − 0·15	+ 0·35 + 0	+ 0·65 + 0·3	+ 0·95 + 0·6	+ 1·25 + 0·9	+ 1·45 + 1·1	+ 1·75 + 1·4	—	+ 2·05 + 1·7	+ 2·35 + 2·0	+ 2·65 + 2·3
1·001– 1·25	− 0·3 − 0·65	− 0 − 0·35	+ 0·2 − 0·15	+ 0·35 + 0	+ 0·65 + 0·3	+ 0·95 + 0·6	+ 1·25 + 0·9	+ 1·45 + 1·1	+ 1·75 + 1·4	+ 2·05 + 1·7	+ 2·35 + 2·0	+ 2·55 + 2·2	+ 2·95 + 2·6
1·251– 1·6	− 0·3 − 0·75	− 0 − 0·45	+ 0·25 − 0·2	+ 0·55 + 0·1	+ 0·85 + 0·4	+ 1·15 + 0·7	+ 1·55 + 1·1	+ 1·85 + 1·4	+ 2·25 + 1·8	+ 2·45 + 2·0	+ 2·85 + 2·4	+ 3·25 + 2·8	+ 3·75 + 3·3
1·601– 2·0	− 0·3 − 0·75	− 0 − 0·45	+ 0·25 − 0·2	+ 0·55 + 0·1	+ 0·85 + 0·4	+ 1·15 + 0·7	+ 1·55 + 1·1	+ 1·85 + 1·4	+ 2·25 + 1·8	+ 2·65 + 2·2	+ 3·25 + 2·8	+ 3·65 + 3·2	+ 4·35 + 3·9
2·001– 2·5	− 0·4 − 0·9	− 0 − 0·5	+ 0·25 − 0·25	+ 0·6 + 0·1	+ 1·0 + 0·5	+ 1·3 + 0·8	+ 1·8 + 1·3	+ 2·1 + 1·6	+ 2·6 + 2·1	+ 3·1 + 2·6	+ 3·9 + 3·4	+ 4·5 + 4·0	+ 5·3 + 4·8
2·501– 3·15	− 0·4 − 0·9	− 0 − 0·5	+ 0·25 − 0·25	+ 0·6 + 0·1	+ 1·0 + 0·5	+ 1·3 + 0·8	+ 1·8 + 1·3	+ 2·2 + 1·7	+ 2·8 + 2·3	+ 3·4 + 2·9	+ 4·5 + 4·0	+ 5·2 + 4·7	+ 6·0 + 5·5
3·151– 4·0	− 0·5 − 1·1	− 0 − 0·6	+ 0·25 − 0·4	+ 0·7 + 0·1	+ 1·1 + 0·5	+ 1·6 + 1·0	+ 2·1 + 1·5	+ 2·6 + 2·0	+ 3·4 + 2·8	+ 4·2 + 3·6	+ 5·5 + 4·9	+ 6·6 + 6·0	+ 7·6 + 7·0
4·001– 5·0	− 0·5 − 1·1	− 0 − 0·6	+ 0·25 − 0·4	+ 0·7 + 0·1	+ 1·1 + 0·5	+ 1·6 + 1·0	+ 2·1 + 1·5	+ 2·8 + 2·2	+ 3·8 + 3·2	+ 4·8 + 4·2	+ 6·6 + 6·0	+ 7·6 + 7·0	+ 9·1 + 8·5
5·001– 5·6	− 0·5 − 1·2	− 0 − 0·7	+ 0·25 − 0·45	+ 0·8 + 0·1	+ 1·3 + 0·6	+ 1·8 + 1·1	+ 2·4 + 1·7	+ 3·2 + 2·5	+ 4·4 + 3·7	+ 5·6 + 4·9	+ 7·7 + 7·0	+ 8·7 + 8·0	+ 10·7 + 10·0
5·601– 6·3	− 0·5 − 1·2	− 0 − 0·7	+ 0·25 − 0·45	+ 0·8 + 0·1	+ 1·3 + 0·6	+ 1·8 + 1·1	+ 2·4 + 1·7	+ 3·3 + 2·6	+ 4·6 + 3·9	+ 6·2 + 5·5	+ 8·2 + 7·5	+ 9·7 + 9·0	+ 11·7 + 11·0
6·301– 7·1	− 0·6 − 1·4	− 0 − 0·8	+ 0·25 − 0·5	+ 0·9 + 0·1	+ 1·4 + 0·6	+ 2·0 + 1·2	+ 2·6 + 1·8	+ 3·6 + 2·8	+ 5·2 + 4·4	+ 6·8 + 6·0	+ 9·3 + 8·5	+ 10·8 + 10·0	+ 12·8 + 12·0
7·101– 8·0	− 0·6 − 1·4	− 0 − 0·8	+ 0·25 − 0·5	+ 0·9 + 0·1	+ 1·4 + 0·6	+ 2·0 + 1·2	+ 2·6 + 1·8	+ 3·7 + 2·9	+ 5·5 + 4·7	+ 7·3 + 6·5	+ 10·3 + 9·5	+ 11·8 + 11·0	+ 14·8 + 14·0
8·001– 9·0	− 0·6 − 1·4	− 0 − 0·8	+ 0·25 − 0·6	+ 0·9 + 0·1	+ 1·5 + 0·7	+ 2·1 + 1·3	+ 2·8 + 2·0	+ 4·1 + 3·3	+ 6·3 + 5·5	+ 7·8 + 7·0	+ 10·8 + 10·0	+ 12·8 + 12·0	+ 15·8 + 15·0
9·001–10·0	− 0·6 − 1·4	− 0 − 0·8	+ 0·25 − 0·6	+ 0·9 + 0·1	+ 1·5 + 0·7	+ 2·1 + 1·3	+ 2·8 + 2·0	+ 4·2 + 3·4	+ 6·3 + 5·5	+ 8·8 + 8·0	+ 11·8 + 11·0	+ 14·8 + 14·0	+ 17·8 + 17·0
10·001–11·2	− 0·7 − 1·6	− 0 − 0·9	+ 0·25 − 0·7	+ 1·1 + 0·2	+ 1·6 + 0·7	+ 2·3 + 1·4	+ 3·1 + 2·2	+ 4·6 + 3·7	+ 7·4 + 6·5	+ 9·4 + 8·5	+ 13·9 + 13·0	+ 15·9 + 15·0	+ 19·9 + 19·0
11·201–12·5	− 0·7 − 1·6	− 0 − 0·9	+ 0·25 − 0·7	+ 1·1 + 0·2	+ 1·6 + 0·7	+ 2·3 + 1·4	+ 3·1 + 2·2	+ 4·7 + 3·8	+ 7·9 + 7·0	+ 10·4 + 9·5	+ 14·9 + 14·0	+ 17·9 + 17·0	+ 21·9 + 21·0
12·501–14·0	− 0·7 − 1·7	− 0 − 1·0	+ 0·3 − 0·7	+ 1·2 + 0·2	+ 1·8 + 0·8	+ 2·5 + 1·5	+ 3·4 + 2·4	+ 5·3 + 4·3	+ 8·5 + 7·5	+ 12·0 + 11·0	+ 16·0 + 15·0	+ 20·0 + 19·0	+ 24·0 + 23·0
14·001–16·0	− 0·7 − 1·7	− 0 − 1·0	+ 0·3 − 0·7	+ 1·2 + 0·2	+ 1·8 + 0·8	+ 2·5 + 1·5	+ 3·4 + 2·4	+ 5·5 + 4·5	+ 9·0 + 8·0	+ 13·0 + 12·0	+ 18·0 + 17·0	+ 22·0 + 21·0	+ 27·0 + 26·0
16·001–18·0	− 0·8 − 1·9	− 0 − 1·1	+ 0·3 − 0·8	+ 1·3 + 0·2	+ 1·9 + 0·8	+ 2·7 + 1·6	+ 3·8 + 2·7	+ 6·1 + 5·0	+ 10·6 + 9·5	+ 14·1 + 13·0	+ 20·1 + 19·0	+ 25·1 + 24·0	+ 31·1 + 30·0
18·001–20·0	− 0·8 − 1·9	− 0 − 1·1	+ 0·3 − 0·8	+ 1·3 + 0·2	+ 1·9 + 0·8	+ 2·7 + 1·6	+ 3·8 + 2·7	+ 6·1 + 5·0	+ 11·1 + 10·0	+ 15·1 + 14·0	+ 22·1 + 21·0	+ 27·1 + 26·0	+ 34·1 + 33·0

Tolerance unit = 0·001 in.

Quality 6

TABLE 24 (*contd.*)

I.S.A. Limits—Shafts: Inch System

Diameter—in.	f6	g6	h6	j6	k6	m6	n6	p6	r6	s6	t6	u6	(v6)	x6	(y6)	z6
0·041– 0·16	− 0·3	− 0·1	− 0	+ 0·2	—	+ 0·35	+ 0·55	+ 0·65	+ 0·75	+ 0·85	—	+ 0·95	—	+ 1·15	—	+ 1·45
	− 0·55	− 0·35	− 0·25	− 0·05		+ 0·1	+ 0·3	+ 0·4	+ 0·5	+ 0·6		+ 0·7		+ 0·9		+ 1·2
0·161–0·315	− 0·4	− 0·2	− 0	+ 0·25	—	+ 0·5	+ 0·7	+ 0·8	+ 1·0	+ 1·2	—	+ 1·2	—	+ 1·4	—	+ 1·8
	− 0·7	− 0·5	− 0·3	− 0·05		+ 0·2	+ 0·4	+ 0·5	+ 0·7	+ 0·9		+ 0·9		+ 1·1		+ 1·5
0·316– 0·4	− 0·6	− 0·2	− 0	+ 0·3	+ 0·4	+ 0·6	+ 0·9	+ 1·0	+ 1·2	+ 1·4	—	+ 1·5	—	+ 1·7	—	+ 2·2
	− 1·0	− 0·6	− 0·4	− 0·1	+ 0	+ 0·2	+ 0·5	+ 0·6	+ 0·8	+ 1·1		+ 1·1		+ 1·3		+ 1·8
0·401– 0·5	− 0·6	− 0·2	− 0	+ 0·3	+ 0·4	+ 0·6	+ 0·9	+ 1·0	+ 1·2	+ 1·4	—	+ 1·7	—	+ 1·9	—	+ 2·4
	− 1·0	− 0·6	− 0·4	− 0·1	+ 0	+ 0·2	+ 0·5	+ 0·6	+ 0·8	+ 1·1		+ 1·3		+ 1·5		+ 2·0
0·501– 0·63	− 0·7	− 0·3	− 0	+ 0·35	+ 0·45	+ 0·7	+ 1·05	+ 1·15	+ 1·35	+ 1·65	—	+ 1·95	+ 1·95	+ 2·15	—	+ 2·65
	− 1·15	− 0·75	− 0·45	− 0·1	+ 0	+ 0·25	+ 0·6	+ 0·7	+ 0·9	+ 1·2		+ 1·5	+ 1·5	+ 1·7		+ 2·2
0·631– 0·8	− 0·7	− 0·3	− 0	+ 0·35	+ 0·45	+ 0·7	+ 1·05	+ 1·15	+ 1·35	+ 1·65	—	+ 1·95	+ 2·15	+ 2·45	+ 2·65	+ 3·05
	− 1·15	− 0·75	− 0·45	− 0·1	+ 0	+ 0·25	+ 0·6	+ 0·7	+ 0·9	+ 1·2		+ 1·5	+ 1·7	+ 2·0	+ 2·2	+ 2·6
0·801– 1·0	− 0·8	− 0·3	− 0	+ 0·35	+ 0·5	+ 0·8	+ 1·1	+ 1·4	+ 1·6	+ 1·9	—	+ 2·2	+ 2·5	+ 2·8	+ 3·1	+ 3·6
	− 1·3	− 0·8	− 0·5	− 0·15	+ 0	+ 0·3	+ 0·6	+ 0·9	+ 1·1	+ 1·4		+ 1·7	+ 2·0	+ 2·3	+ 2·6	+ 3·1
1·001– 1·25	− 0·8	− 0·3	− 0	+ 0·35	+ 0·5	+ 0·8	+ 1·1	+ 1·4	+ 1·6	+ 1·9	+ 2·2	+ 2·5	+ 2·7	+ 3·1	+ 3·6	+ 4·1
	− 1·3	− 0·8	− 0·5	− 0·15	+ 0	+ 0·3	+ 0·6	+ 0·9	+ 1·1	+ 1·4	+ 1·7	+ 2·0	+ 2·2	+ 2·6	+ 3·1	+ 3·6
1·251 – 1·6	− 1·0	− 0·3	− 0	+ 0·4	+ 0·7	+ 1·0	+ 1·3	+ 1·7	+ 2·0	+ 2·4	+ 2·6	+ 3·0	+ 3·4	+ 3·9	+ 4·4	+ 5·1
	− 1·6	− 0·9	− 0·6	− 0·2	+ 0·1	+ 0·4	+ 0·7	+ 1·1	+ 1·4	+ 1·8	+ 2·0	+ 2·4	+ 2·8	+ 3·3	+ 3·8	+ 4·5
1·601– 2·0	− 1·0	− 0·3	− 0	+ 0·4	+ 0·7	+ 1·0	+ 1·3	+ 1·7	+ 2·0	+ 2·4	+ 2·8	+ 3·4	+ 3·8	+ 4·5	+ 5·2	+ 6·1
	− 1·6	− 0·9	− 0·6	− 0·2	+ 0·1	+ 0·4	+ 0·7	+ 1·1	+ 1·4	+ 1·8	+ 2·2	+ 2·8	+ 3·2	+ 3·9	+ 4·6	+ 5·5
2·001– 2·5	− 1·2	− 0·4	− 0	+ 0·45	+ 0·8	+ 1·2	+ 1·5	+ 2·0	+ 2·3	+ 2·8	+ 3·3	+ 4·1	+ 4·7	+ 5·5	+ 6·2	+ 7·7
	− 1·9	− 1·1	− 0·7	− 0·25	+ 0·1	+ 0·5	+ 0·8	+ 1·3	+ 1·6	+ 2·1	+ 2·6	+ 3·4	+ 4·0	+ 4·8	+ 5·5	+ 7·0
2·501– 3·15	− 1·2	− 0·4	− 0	+ 0·45	+ 0·8	+ 1·2	+ 1·5	+ 2·0	+ 2·4	+ 3·0	+ 3·6	+ 4·7	+ 5·4	+ 6·2	+ 7·7	+ 8·7
	− 1·9	− 1·1	− 0·7	− 0·25	+ 0·1	+ 0·5	+ 0·8	+ 1·3	+ 1·7	+ 2·3	+ 2·9	+ 4·0	+ 4·7	+ 5·5	+ 7·0	+ 8·0
3·151– 4·0	− 1·4	− 0·5	− 0	+ 0·5	+ 1·0	+ 1·4	+ 1·9	+ 2·4	+ 2·9	+ 3·7	+ 4·5	+ 5·8	+ 6·9	+ 7·9	+ 9·4	+ 10·9
	− 2·3	− 1·4	− 0·9	− 0·4	+ 0·1	+ 0·5	+ 1·0	+ 1·5	+ 2·0	+ 2·8	+ 3·6	+ 4·9	+ 6·0	+ 7·0	+ 8·5	+ 10·0
4·001– 5·0	− 1·4	− 0·5	− 0	+ 0·5	+ 1·0	+ 1·4	+ 1·9	+ 2·4	+ 3·1	+ 4·1	+ 5·1	+ 6·9	+ 7·9	+ 9·4	+ 10·9	+ 13·9
	− 2·3	− 1·4	− 0·9	− 0·4	+ 0·1	+ 0·5	+ 1·0	+ 1·5	+ 2·2	+ 3·2	+ 4·2	+ 6·0	+ 7·0	+ 8·5	+ 10·0	+ 13·0
5·001– 5·6	− 1·7	− 0·5	− 0	+ 0·6	+ 1·1	+ 1·6	+ 2·1	+ 2·7	+ 3·5	+ 4·7	+ 5·9	+ 8·0	+ 9·0	+ 11·0	+ 13·0	+ 16·0
	− 2·7	− 1·5	− 1·0	− 0·45	+ 0·1	+ 0·6	+ 1·1	+ 1·7	+ 2·5	+ 3·7	+ 4·9	+ 7·0	+ 8·0	+ 10·0	+ 12·0	+ 15·0
5·601– 6·3	− 1·7	− 0·5	− 0	+ 0·6	+ 1·1	+ 1·6	+ 2·1	+ 2·7	+ 3·6	+ 4·9	+ 6·5	+ 8·5	+ 10·0	+ 12·0	+ 14·0	+ 17·0
	− 2·7	− 1·5	− 1·0	− 0·45	+ 0·1	+ 0·6	+ 1·1	+ 1·7	+ 2·6	+ 3·9	+ 5·5	+ 7·5	+ 9·0	+ 11·0	+ 13·0	+ 16·0
6·301– 7·1	− 1·8	− 0·6	− 0	+ 0·6	+ 1·2	+ 1·7	+ 2·3	+ 2·9	+ 3·9	+ 5·5	+ 7·1	+ 9·6	+ 11·1	+ 13·1	+ 16·1	+ 19·1
	− 2·9	− 1·7	− 1·1	− 0·5	+ 0·1	+ 0·6	+ 1·2	+ 1·8	+ 2·8	+ 4·4	+ 6·0	+ 8·5	+ 10·0	+ 12·0	+ 15·0	+ 18·0
7·101– 8·0	− 1·8	− 0·6	− 0	+ 0·6	+ 1·2	+ 1·7	+ 2·3	+ 2·9	+ 4·0	+ 5·8	+ 7·6	+ 10·6	+ 12·1	+ 15·1	+ 18·1	+ 22·1
	− 2·9	− 1·7	− 1·1	− 0·5	+ 0·1	+ 0·6	+ 1·2	+ 1·8	+ 2·9	+ 4·7	+ 6·5	+ 9·5	+ 11·0	+ 14·0	+ 17·0	+ 21·0
8·001– 9·0	− 2·0	− 0·6	− 0	+ 0·6	+ 1·3	+ 1·9	+ 2·5	+ 3·2	+ 4·5	+ 6·7	+ 8·2	+ 11·2	+ 13·2	+ 16·2	+ 20·2	+ 24·2
	− 3·2	− 1·8	− 1·2	− 0·6	+ 0·1	+ 0·7	+ 1·3	+ 2·0	+ 3·3	+ 5·5	+ 7·0	+ 10·0	+ 12·0	+ 15·0	+ 19·0	+ 23·0
9·001–10·0	− 2·0	− 0·6	− 0	+ 0·6	+ 1·3	+ 1·9	+ 2·5	+ 3·2	+ 4·6	+ 6·7	+ 9·2	+ 12·2	+ 15·2	+ 18·2	+ 22·2	+ 27·2
	− 3·2	− 1·8	− 1·2	− 0·6	+ 0·1	+ 0·7	+ 1·3	+ 2·0	+ 3·4	+ 5·5	+ 8·0	+ 11·0	+ 14·0	+ 17·0	+ 21·0	+ 26·0
10·001–11·2	− 2·2	− 0·7	− 0	+ 0·6	+ 1·5	+ 2·0	+ 2·7	+ 3·5	+ 5·0	+ 7·8	+ 9·8	+ 14·3	+ 16·3	+ 20·3	+ 24·3	+ 29·3
	− 3·5	− 2·0	− 1·3	− 0·7	+ 0·2	+ 0·7	+ 1·4	+ 2·2	+ 3·7	+ 6·5	+ 8·5	+ 13·0	+ 15·0	+ 19·0	+ 23·0	+ 28·0
11·201–12·5	− 2·2	− 0·7	− 0	+ 0·6	+ 1·5	+ 2·0	+ 2·7	+ 3·5	+ 5·1	+ 8·3	+ 10·8	+ 15·3	+ 18·3	+ 22·3	+ 27·3	+ 33·3
	− 3·5	− 2·0	− 1·3	− 0·7	+ 0·2	+ 0·7	+ 1·4	+ 2·2	+ 3·8	+ 7·0	+ 9·5	+ 14·0	+ 17·0	+ 21·0	+ 26·0	+ 32·0
12·501–14·0	− 2·4	− 0·7	− 0	+ 0·7	+ 1·6	+ 2·2	+ 2·9	+ 3·8	+ 5·7	+ 8·9	+ 12·4	+ 16·4	+ 20·4	+ 24·4	+ 30·4	+ 36·4
	− 3·8	− 2·1	− 1·4	− 0·7	+ 0·2	+ 0·8	+ 1·5	+ 2·4	+ 4·3	+ 7·5	+ 11·0	+ 15·0	+ 19·0	+ 23·0	+ 29·0	+ 35·0
14·001–16·0	− 2·4	− 0·7	− 0	+ 0·7	+ 1·6	+ 2·2	+ 2·9	+ 3·8	+ 5·9	+ 9·4	+ 13·4	+ 18·4	+ 22·4	+ 27·4	+ 33·4	+ 41·4
	− 3·8	− 2·1	− 1·4	− 0·7	+ 0·2	+ 0·8	+ 1·5	2·4	+ 4·5	+ 8·0	+ 12·0	+ 17·0	+ 21·0	+ 26·0	+ 32·0	+ 40·0
16·001–18·0	− 2·7	− 0·8	− 0	+ 0·7	+ 1·7	+ 2·3	+ 3·1	+ 4·2	+ 6·5	+ 11·0	+ 14·5	+ 20·5	+ 25·5	+ 31·5	+ 37·5	+ 46·5
	− 4·2	− 2·3	− 1·5	− 0·8	+ 0·2	+ 0·8	+ 1·6	+ 2·7	+ 5·0	+ 9·5	+ 13·0	+ 19·0	+ 24·0	+ 30·0	+ 36·0	+ 45·0
18·001–20·0	− 2·7	− 0·8	− 0	+ 0·7	+ 1·7	+ 2·3	+ 3·1	+ 4·2	+ 6·5	+ 11·5	+ 15·5	+ 22·5	+ 27·5	+ 34·5	+ 41·5	+ 51·5
	− 4·2	− 2·3	− 1·5	− 0·8	+ 0·2	+ 0·8	+ 1·6	+ 2·7	+ 5·0	+ 10·0	+ 14·0	+ 21·0	+ 26·0	+ 33·0	+ 40·0	+ 50·0

Tolerance unit = 0·001 in.

TABLE 24 (contd.)

Quality 7

A. Limits—Shafts: Inch System

Diameter—in.	e7	f7	h7	j7	k7	m7	n7	p7	r7	s7	t7	u7	(v7)	x7	(y7)	z7
0·041– 0·16	− 0·6	− 0·3	− 0	+ 0·25	—	—	+ 0·65	+ 0·75	+ 0·85	+ 0·95	—	+ 1·05	—	+ 1·25	—	+ 1·55
	− 0·95	− 0·65	− 0·35	− 0·1			+ 0·3	+ 0·4	+ 0·5	+ 0·6		+ 0·7		+ 0·9		+ 1·2
0·161– 0·315	− 0·9	− 0·4	− 0	+ 0·35	—	—	+ 0·9	+ 1·0	+ 1·2	+ 1·4	—	+ 1·4	—	+ 1·6	—	+ 2·0
	− 1·4	− 0·9	− 0·5	− 0·15			+ 0·4	+ 0·5	+ 0·7	+ 0·9		+ 0·9		+ 1·1		+ 1·5
0·316– 0·4	− 1·1	− 0·6	− 0	+ 0·4	+ 0·6	+ 0·8	+ 1·1	+ 1·2	+ 1·4	+ 1·6	—	+ 1·7	—	+ 1·9	—	+ 2·4
	− 1·7	− 1·2	− 0·6	− 0·2	+ 0	+ 0·2	+ 0·5	+ 0·6	+ 0·8	+ 1·1		+ 1·1		+ 1·3		+ 1·8
0·401– 0·5	− 1·1	− 0·6	− 0	+ 0·4	+ 0·6	+ 0·8	+ 1·1	+ 1·2	+ 1·4	+ 1·6	—	+ 1·9	—	+ 2·1	—	+ 2·6
	− 1·7	− 1·2	− 0·6	− 0·2	+ 0	+ 0·2	+ 0·5	+ 0·6	+ 0·8	+ 1·1		+ 1·3		+ 1·5		+ 2·0
0·501– 0·63	− 1·4	− 0·7	− 0	+ 0·4	+ 0·7	+ 0·95	+ 1·3	+ 1·4	+ 1·6	+ 1·9	—	+ 2·2	+ 2·2	+ 2·4	—	+ 2·9
	− 2·1	− 1·4	− 0·7	− 0·3	+ 0	+ 0·25	+ 0·6	+ 0·7	+ 0·9	+ 1·2		+ 1·5	+ 1·5	+ 1·7		+ 2·2
0·631– 0·8	− 1·4	− 0·7	− 0	+ 0·4	+ 0·7	+ 0·95	+ 1·3	+ 1·4	+ 1·6	+ 1·9	—	+ 2·2	+ 2·4	+ 2·7	+ 2·9	+ 3·3
	− 2·1	− 1·4	− 0·7	− 0·3	+ 0	+ 0·2	+ 0·6	+ 0·7	+ 0·9	+ 1·2		+ 1·5	+ 1·7	+ 2·0	+ 2·2	+ 2·6
0·801– 1·0	− 1·6	− 0·8	− 0	+ 0·5	+ 0·8	+ 1·1	+ 1·4	+ 1·7	+ 1·9	+ 2·2	—	+ 2·5	+ 2·8	+ 3·1	+ 3·4	+ 3·9
	− 2·4	− 1·6	− 0·8	− 0·3	+ 0	+ 0·3	+ 0·6	+ 0·9	+ 1·1	+ 1·4		+ 1·7	+ 2·0	+ 2·3	+ 2·6	+ 3·1
1·001– 1·25	− 1·6	− 0·8	− 0	+ 0·5	+ 0·8	+ 1·1	+ 1·4	+ 1·7	+ 1·9	+ 2·2	+ 2·5	+ 2·8	+ 3·0	+ 3·4	+ 3·9	+ 4·4
	− 2·4	− 1·6	− 0·8	− 0·3	+ 0	+ 0·3	+ 0·6	+ 0·9	+ 1·1	+ 1·4	+ 1·7	+ 2·0	+ 2·2	+ 2·6	+ 3·1	+ 3·6
1·251– 1·6	− 2·0	− 1·0	− 0	+ 0·6	+ 1·1	+ 1·4	+ 1·7	+ 2·1	+ 2·4	+ 2·8	+ 3·0	+ 3·4	+ 3·8	+ 4·3	+ 4·8	+ 5·5
	− 3·0	− 2·0	− 1·0	− 0·4	+ 0·1	+ 0·4	+ 0·7	+ 1·1	+ 1·4	+ 1·8	+ 2·0	+ 2·4	+ 2·8	+ 3·3	+ 3·8	+ 4·5
1·601– 2·0	− 2·0	− 1·0	− 0	+ 0·6	+ 1·1	+ 1·4	+ 1·7	+ 2·1	+ 2·4	+ 2·8	+ 3·2	+ 3·8	+ 4·2	+ 4·9	+ 5·6	+ 6·5
	− 3·0	− 2·0	− 1·0	− 0·4	+ 0·1	+ 0·4	+ 0·7	+ 1·1	+ 1·4	+ 1·8	+ 2·2	+ 2·8	+ 3·2	+ 3·9	+ 4·6	+ 5·5
2·001– 2·5	− 2·4	− 1·2	− 0	+ 0·7	+ 1·3	+ 1·7	+ 2·0	+ 2·5	+ 2·8	+ 3·3	+ 3·8	+ 4·6	+ 5·2	+ 6·0	+ 6·7	+ 8·2
	− 3·6	− 2·4	− 1·2	− 0·5	+ 0·1	+ 0·5	+ 0·8	+ 1·3	+ 1·6	+ 2·1	+ 2·6	+ 3·4	+ 4·0	+ 4·8	+ 5·5	+ 7·0
2·501– 3·15	− 2·4	− 1·2	− 0	+ 0·7	+ 1·3	+ 1·7	+ 2·0	+ 2·5	+ 2·9	+ 3·5	+ 4·1	+ 5·2	+ 5·9	+ 6·7	+ 8·2	+ 9·2
	− 3·6	− 2·4	− 1·2	− 0·5	+ 0·1	+ 0·5	+ 0·8	+ 1·3	+ 1·7	+ 2·3	+ 2·9	+ 4·0	+ 4·7	+ 5·5	+ 7·0	+ 8·0
3·151– 4·0	− 2·9	− 1·4	− 0	+ 0·8	+ 1·5	+ 1·9	+ 2·4	+ 2·9	+ 3·4	+ 4·2	+ 5·0	+ 6·3	+ 7·4	+ 8·4	+ 9·9	+ 11·4
	− 4·3	− 2·8	− 1·4	− 0·6	+ 0·1	+ 0·5	+ 1·0	+ 1·5	+ 2·0	+ 2·8	+ 3·6	+ 4·9	+ 6·0	+ 7·0	+ 8·5	+ 10·0
4·001– 5·0	− 2·9	− 1·4	− 0	+ 0·8	+ 1·5	+ 1·9	+ 2·4	+ 2·9	+ 3·6	+ 4·6	+ 5·6	+ 7·4	+ 8·4	+ 9·9	+ 11·4	+ 14·4
	− 4·3	− 2·8	− 1·4	− 0·6	+ 0·1	+ 0·5	+ 1·0	+ 1·5	+ 2·2	+ 3·2	+ 4·2	+ 6·0	+ 7·0	+ 8·5	+ 10·0	+ 13·0
5·001– 5·6	− 3·3	− 1·7	− 0	+ 0·9	+ 1·7	+ 2·2	+ 2·7	+ 3·3	+ 4·1	+ 5·3	+ 6·5	+ 8·6	+ 9·6	+ 11·6	+ 13·6	+ 16·6
	− 4·9	− 3·3	− 1·6	− 0·7	+ 0·1	+ 0·6	+ 1·1	+ 1·7	+ 2·5	+ 3·7	+ 4·9	+ 7·0	+ 8·0	+ 10·0	+ 12·0	+ 15·0
5·601– 6·3	− 3·3	− 1·7	− 0	+ 0·9	+ 1·7	+ 2·2	+ 2·7	+ 3·3	+ 4·2	+ 5·5	+ 7·1	+ 9·1	+ 10·6	+ 12·6	+ 14·6	+ 17·6
	− 4·9	− 3·3	− 1·6	− 0·7	+ 0·1	+ 0·6	+ 1·1	+ 1·7	+ 2·6	+ 3·9	+ 5·5	+ 7·5	+ 9·0	+ 11·0	+ 13·0	+ 16·0
6 301– 7·1	− 3·6	− 1·8	− 0	+ 0·9	+ 1·8	+ 2·3	+ 2·9	+ 3·5	+ 4·5	+ 6·1	+ 7·7	+ 10·2	+ 11·7	+ 13·7	+ 16·7	+ 19·7
	− 5·3	− 3·5	− 1·7	− 0·8	+ 0·1	+ 0·6	+ 1·2	+ 1·8	+ 2·8	+ 4·4	+ 6·0	+ 8·5	+ 10·0	+ 12·0	+ 15·0	+ 18·0
7·101– 8·0	− 3·6	− 1·8	− 0	+ 0·9	+ 1·8	+ 2·3	+ 2·9	+ 3·5	+ 4·6	+ 6·4	+ 8·2	+ 11·2	+ 12·7	+ 15·7	+ 18·7	+ 22·7
	− 5·3	− 3·5	− 1·7	− 0·8	+ 0·1	+ 0·6	+ 1·2	+ 1·8	+ 2·9	+ 4·7	+ 6·5	+ 9·5	+ 11·0	+ 14·0	+ 17·0	+ 21·0
8·001– 9·0	− 4·0	− 2·0	− 0	+ 1·0	+ 2·0	+ 2·6	+ 3·2	+ 3·9	+ 5·2	+ 7·4	+ 8·9	+ 11·9	+ 13·9	+ 16·9	+ 20·9	+ 24·9
	− 5·9	− 3·9	− 1·9	− 0·9	+ 0·1	+ 0·7	+ 1·3	+ 2·0	+ 3·3	+ 5·5	+ 7·0	+ 10·0	+ 12·0	+ 15·0	+ 19·0	+ 23·0
9·001–10·0	− 4·0	− 2·0	− 0	+ 1·0	+ 2 0	+ 2·6	+ 3·2	+ 3·9	+ 5·3	+ 7·4	+ 9·9	+ 12 9	+ 15·9	+ 18·9	+ 22·9	+ 27·9
	− 5·9	− 3·9	− 1 9	− 0·9	+ 0·1	+ 0·7	+ 1·3	+ 2·0	+ 3·4	+ 5·5	+ 8·0	+ 11·0	+ 14·0	+ 17·0	+ 21·0	+ 26·0
10·001–11·2	− 4·4	− 2·2	− 0	+ 1·0	+ 2·2	+ 2·7	+ 3·4	+ 4·2	+ 5·7	+ 8·5	+ 10·5	+ 15·0	+ 17·0	+ 21·0	+ 25·0	+ 30·0
	− 6·4	− 4·2	− 2·0	− 1·0	+ 0·2	+ 0·7	+ 1·4	+ 2·2	+ 3·7	+ 6·5	+ 8·5	+ 13·0	+ 15·0	+ 19·0	+ 23 0	+ 28·0
11·201–12·5	− 4·4	− 2·2	− 0	+ 1·0	+ 2·2	+ 2·7	+ 3·4	+ 4·2	+ 5·8	+ 9·0	+ 11·5	+ 16·0	+ 19·0	+ 23·0	+ 28·0	+ 34·0
	− 6·4	− 4·2	− 2·0	− 1·0	+ 0·2	+ 0·7	+ 1·4	+ 2·2	+ 3·8	+ 7·0	+ 9·5	+ 14 0	+ 17·0	+ 21·0	+ 26·0	+ 32·0
12 501–14·0	− 4·8	− 2·4	− 0	+ 1·1	+ 2·4	+ 3·0	+ 3·7	+ 4·6	+ 6·5	+ 9·7	+ 13·2	+ 17·2	+ 21·2	+ 25·2	+ 31·2	+ 37·2
	− 7·0	− 4·6	− 2·2	− 1·1	+ 0·2	+ 0·8	+ 1·5	+ 2·4	+ 4·3	+ 7·5	+ 11·0	+ 15·0	+ 19·0	+ 23·0	+ 29·0	+ 35·0
14·001–16·0	− 4·8	− 2·4	− 0	+ 1·1	+ 2·4	+ 3·0	+ 3·7	+ 4·6	+ 6·7	+ 10·2	+ 14·2	+ 19·2	+ 23·2	+ 28·2	+ 34·2	+ 42·2
	− 7·0	− 4·6	− 2·2	− 1·1	+ 0·2	+ 0·8	+ 1·5	+ 2·4	+ 4·5	+ 8·0	+ 12·0	+ 17·0	+ 21·0	+ 26·0	+ 32·0	+ 40·0
16·001–18·0	− 5·5	− 2·7	− 0	+ 1·3	+ 2·7	+ 3·3	+ 4·1	+ 5·2	+ 7·5	+ 12·0	+ 15·5	+ 21·5	+ 26·5	+ 32·5	+ 38·5	+ 47·5
	− 8·0	− 5·2	− 2·5	− 1·2	+ 0·2	+ 0·8	+ 1·6	+ 2·7	+ 5·0	+ 9·5	+ 13·0	+ 19·0	+ 24·0	+ 30·0	+ 36·0	+ 45·0
18·001–20·0	− 5·5	− 2·7	− 0	+ 1·3	+ 2·7	+ 3·3	+ 4·1	+ 5·2	+ 7·5	+ 12·5	+ 16·5	+ 23·5	+ 28·5	+ 35·5	+ 42·5	+ 52·5
	− 8·0	− 5·2	− 2·5	− 1·2	+ 0·2	+ 0·8	+ 1·6	+ 2·7	+ 5·0	+ 10·0	+ 14·0	+ 21·0	+ 26·0	+ 33·0	+ 40·0	+ 50·0

Tolerance unit = 0·001 in.

Qualities 8 and 9

TABLE 24 (*contd.*)
I.S.A. Limits—Shafts: Inch System

Diameter—in.	b8	c8	d8	e8	f8	h8	j8	k8	a9	b9	c9	d9	e9	h9	j9	k9
0·041– 0·16	− 5·5	− 2·4	− 0·9	− 0·6	− 0·3	− 0	+ 0·3	+ 0·6	− 11·0	− 5·5	− 2·4	− 0·9	− 0·6	− 0	+ 0·5	+ 0·9
	− 6·1	− 3·0	− 1·5	− 1·2	− 0·9	− 0·6	− 0·3	+ 0	− 11·9	− 6·4	− 3·3	− 1·8	− 1·5	− 0·9	− 0·4	+ 0
0·161– 0·315	− 5·5	− 2·9	− 1·4	− 0·9	− 0·4	− 0	+ 0·4	+ 0·8	− 11·0	− 5·5	− 2·9	− 1·4	− 0·9	− 0	+ 0·7	+ 1·3
	− 6·3	− 3·7	− 2·2	− 1·7	− 1·2	− 0·8	− 0·4	+ 0	− 12·3	− 6·8	− 4·2	− 2·7	− 2·2	− 1·3	− 0·6	+ 0
0·316– 0·4	− 6·0	− 3·2	− 1·7	− 1·1	− 0·6	− 0	+ 0·5	+ 1·0	− 11·0	− 6·0	− 3·2	− 1·7	− 1·1	− 0	+ 0·8	+ 1·5
	− 7·0	− 4·2	− 2·7	− 2·1	− 1·6	− 1·0	− 0·5	+ 0	− 12·5	− 7·5	− 4·7	− 3·2	− 2·6	− 1·5	− 0·7	+ 0
0·401– 0·5	− 6·0	− 3·2	− 1·7	− 1·1	− 0·6	− 0	+ 0·5	+ 1·0	− 11·0	− 6·0	− 3·2	− 1·7	− 1·1	− 0	+ 0·8	+ 1·5
	− 7·0	− 4·2	− 2·7	− 2·1	− 1·6	− 1·0	− 0·5	+ 0	− 12·5	− 7·5	− 4·7	− 3·2	− 2·6	− 1·5	− 0·7	+ 0
0·501– 0·63	− 6·0	− 3·6	− 2·1	− 1·4	− 0·7	− 0	+ 0·6	+ 1·1	− 11·0	− 6·0	− 3·6	− 2·1	− 1·4	− 0	+ 0·9	+ 1·8
	− 7·1	− 4·7	− 3·2	− 2·5	− 1·8	− 1·1	− 0·5	+ 0	− 12·8	− 7·8	− 5·4	− 3·9	− 3·2	− 1·8	− 0·9	+ 0
0·631– 0·8	− 6·0	− 3·6	− 2·1	− 1·4	− 0·7	− 0	+ 0·6	+ 1·1	− 11·0	− 6·0	− 3·6	− 2·1	− 1·4	− 0	+ 0·9	+ 1·8
	− 7·1	− 4·7	− 3·2	− 2·5	− 1·8	− 1·1	− 0·5	+ 0	− 12·8	− 7·8	− 5·4	− 3·9	− 3·2	− 1·8	− 0·9	+ 0
0·801– 1·0	− 6·5	− 3·9	− 2·6	− 1·6	− 0·8	− 0	+ 0·7	+ 1·3	− 12·0	− 6·5	− 3·9	− 2·6	− 1·6	− 0	+ 1·1	+ 2·1
	− 7·8	− 5·2	− 3·9	− 2·9	− 2·1	− 1·3	− 0·6	+ 0	− 14·1	− 8·6	− 6·0	− 4·7	− 3·7	− 2·1	− 1·0	+ 0
1·001– 1·25	− 6·5	− 3·9	− 2·6	− 1·6	− 0·8	− 0	+ 0·7	+ 1·3	− 12·0	− 6·5	− 3·9	− 2·6	− 1·6	− 0	+ 1·1	+ 2·1
	− 7·8	− 5·2	− 3·9	− 2·9	− 2·1	− 1·3	− 0·6	+ 0	− 14·1	− 8·6	− 6·0	− 4·7	− 3·7	− 2·1	− 1·0	+ 0
1·251– 1·6	− 6·5	− 4·2	− 3·2	− 2·0	− 1·0	− 0	+ 0·8	+ 1·6	− 12·0	− 6·5	− 4·2	− 3·2	− 2·0	− 0	+ 1·3	+ 2·5
	− 8·1	− 5·8	− 4·8	− 3·6	− 2·6	− 1·6	− 0·8	+ 0	− 14·5	− 9·0	− 6·7	− 5·7	− 4·5	− 2·5	− 1·2	+ 0
1·601– 2·0	− 7·0	− 4·4	− 3·2	− 2·0	− 1·0	− 0	+ 0·8	+ 1·6	− 13·0	− 7·0	− 4·4	− 3·2	− 2·0	− 0	+ 1·3	+ 2·5
	− 8·6	− 6·0	− 4·8	− 3·6	− 2·6	− 1·6	− 0·8	+ 0	− 15·5	− 9·5	− 6·9	− 5·7	− 4·5	− 2·5	− 1·2	+ 0
2·001– 2·5	− 7·5	− 5·5	− 3·9	− 2·4	− 1·2	− 0	+ 0·9	+ 1·8	− 13·0	− 7·5	− 5·5	− 3·9	− 2·4	− 0	+ 1·5	+ 2·9
	− 9·3	− 7·3	− 5·7	− 4·2	− 3·0	− 1·8	− 0·9	+ 0	− 15·9	− 10·4	− 8·4	− 6·8	− 5·3	− 2·9	− 1·4	+ 0
2·501– 3·15	− 8·0	− 6·0	− 3·9	− 2·4	− 1·2	− 0	+ 0·9	+ 1·8	− 14·0	− 8·0	− 6·0	− 3·9	− 2·4	− 0	+ 1·5	+ 2·9
	− 9·8	− 7·8	− 5·7	− 4·2	− 3·0	− 1·8	− 0·9	+ 0	− 16·9	− 10·9	− 8·9	− 6·8	− 5·3	− 2·9	− 1·4	+ 0
3·151– 4·0	− 8·5	− 6·5	− 4·8	− 2·9	− 1·4	− 0	+ 1·1	+ 2·2	− 15·0	− 8·5	− 6·5	− 4·8	− 2·9	− 0	+ 1·8	+ 3·5
	− 10·7	− 8·7	− 7·0	− 5·1	− 3·6	− 2·2	− 1·1	+ 0	− 18·5	− 12·0	− 10·0	− 8·3	− 6·4	− 3·5	− 1·7	+ 0
4·001– 5·0	− 9·5	− 7·5	− 4·8	− 2·9	− 1·4	− 0	+ 1·1	+ 2·2	− 16·0	− 9·5	− 7·5	− 4·8	− 2·9	− 0	+ 1·8	+ 3·5
	− 11·7	− 9·7	− 7·0	− 5·1	− 3·6	− 2·2	− 1·1	+ 0	− 19·5	− 13·0	− 11·0	− 8·3	− 6·4	− 3·5	− 1·7	+ 0
5·001– 5·6	− 9·5	− 8·0	− 5·5	− 3·3	− 1·7	− 0	+ 1·3	+ 2·5	− 19·0	− 9·5	− 8·0	− 5·5	− 3·3	− 0	+ 2·0	+ 3·9
	− 12·0	− 10·5	− 8·0	− 5·8	− 4·2	− 2·5	− 1·2	+ 0	− 22·9	− 13·4	− 11·9	− 9·4	− 7·2	− 3·9	− 1·9	+ 0
5·601– 6·3	− 11·0	− 8·5	− 5·5	− 3·3	− 1·7	− 0	+ 1·3	+ 2·5	− 21·0	− 11·0	− 8·5	− 5·5	− 3·3	− 0	+ 2·0	+ 3·9
	− 13·5	− 11·0	− 8·0	− 5·8	− 4·2	− 2·5	− 1·2	+ 0	− 24·9	− 14·9	− 12·4	− 9·4	− 7·2	− 3·9	− 1·9	+ 0
6·301– 7·1	− 12·0	− 9·0	− 6·0	− 3·6	− 1·8	− 0	+ 1·4	+ 2·7	− 23·0	− 12·0	− 9·0	− 6·0	− 3·6	− 0	+ 2·2	+ 4·3
	− 14·7	− 11·7	− 8·7	− 6·3	− 4·5	− 2·7	− 1·3	+ 0	− 27·3	− 16·3	− 13·3	− 10·3	− 7·9	− 4·3	− 2·1	+ 0
7·101– 8·0	− 14·0	− 10·0	− 6·0	− 3·6	− 1·8	− 0	+ 1·4	+ 2·7	− 26·0	− 14·0	− 10·0	− 6·0	− 3·6	− 0	+ 2·2	+ 4·3
	− 16·7	− 12·7	− 8·7	− 6·3	− 4·5	− 2·7	− 1·3	+ 0	− 30·3	− 18·3	− 14·3	− 10·3	− 7·9	− 4·3	− 2·1	+ 0
8·001– 9·0	− 15·0	− 11·0	− 7·0	− 4·0	− 2·0	− 0	+ 1·5	+ 2·9	− 30·0	− 15·0	− 11·0	− 7·0	− 4·0	− 0	+ 2·4	+ 4·7
	− 17·9	− 13·9	− 9·9	− 6·9	− 4·9	− 2·9	− 1·4	+ 0	− 34·7	− 19·7	− 15·7	− 11·7	− 8·7	− 4·7	− 2·3	+ 0
9·001–10·0	− 17·0	− 11·0	− 7·0	− 4·0	− 2·0	− 0	+ 1·5	+ 2·9	− 33·0	− 17·0	− 11·0	− 7·0	− 4·0	− 0	+ 2·4	+ 4·7
	− 19·9	− 13·9	− 9·9	− 6·9	− 4·9	− 2·9	− 1·4	+ 0	− 37·7	− 21·7	− 15·7	− 11·7	− 8·7	− 4·7	− 2·3	+ 0
10·001–11·2	− 19·0	− 12·0	− 7·5	− 4·4	− 2·2	− 0	+ 1·6	+ 3·2	− 37·0	− 19·0	− 12·0	− 7·5	− 4·4	− 0	+ 2·5	+ 5·0
	− 22·2	− 15·2	− 10·7	− 7·6	− 5·4	− 3·2	− 1·6	+ 0	− 42·0	− 24·0	− 17·0	− 12·5	− 9·4	− 5·0	− 2·5	+ 0
11·201–12·5	− 21·0	− 13·0	− 7·5	− 4·4	− 2·2	− 0	+ 1·6	+ 3·2	− 41·0	− 21·0	− 13·0	− 7·5	− 4·4	− 0	+ 2·5	+ 5·0
	− 24·2	− 16·2	− 10·7	− 7·6	− 5·4	− 3·2	− 1·6	+ 0	− 46·0	− 26·0	− 18·0	− 12·5	− 9·4	− 5·0	− 2·5	+ 0
12·501–14·0	− 24·0	− 14·0	− 8·5	− 4·8	− 2·4	− 0	+ 1·8	+ 3·5	− 46·0	− 24·0	− 14·0	− 8·5	− 4·8	− 0	+ 2·8	+ 5·5
	− 27·5	− 17·5	− 12·0	− 8·3	− 5·9	− 3·5	− 1·7	+ 0	− 51·5	− 29·5	− 19·5	− 14·0	− 10·3	− 5·5	− 2·7	+ 0
14·001–16·0	− 27·0	− 16·0	− 8·5	− 4·8	− 2·4	− 0	+ 1·8	+ 3·5	− 52·0	− 27·0	− 16·0	− 8·5	− 4·8	− 0	+ 2·8	+ 5·5
	− 30·5	− 19·5	− 12·0	− 8·3	− 5·9	− 3·5	− 1·7	+ 0	− 57·5	− 32·5	− 21·5	− 14·0	− 10·3	− 5·5	− 2·7	+ 0
16·001–18·0	− 31·0	− 17·0	− 9·5	− 5·5	− 2·7	− 0	[illegible]	+ 3·9	− 59·0	− 31·0	− 17·0	− 9·5	− 5·5	− 0	+ 3·0	+ 6·0
	− 34·9	− 20·9	− 13·4	− 9·4	− 6·6	− 3·9	− 1·9	+ 0	− 65·0	− 37·0	− 23·0	− 15·5	− 11·5	− 6·0	− 3·0	+ 0
18·001–20·0	− 34·0	− 19·0	− 9·5	− 5·5	− 2·7	− 0	+ 2·0	+ 3·9	− 66·0	− 34·0	− 19·0	− 9·5	− 5·5	− 0	+ 3·0	+ 6·0
	− 37·9	− 22·9	− 13·4	− 9·4	− 6·6	− 3·9	− 1·9	+ 0	− 72·0	− 40·0	− 25·0	− 15·5	− 11·5	− 6·0	− 3·0	+ 0

Tolerance unit = 0·001 in.

TABLE 24 (*contd.*)

I.S.A. LIMITS—SHAFTS: INCH SYSTEM

Qualities 10 and 11

Diameter —in.	d10	h10	j10	k10	a11	b11	c11	d11	h11	j11	k11
0·041– 0·16	− 0·9 − 2·3	− 0 − 1·4	+ 0·7 − 0·7	+ 1·4 + 0	− 11·0 − 13·3	− 5·5 − 7·8	− 2·4 − 4·7	− 0·9 − 3·2	− 0 − 2·3	+ 1·2 − 1·1	+ 2·3 + 0
0·161– 0·315	− 1·4 − 3·4	− 0 − 2·0	+ 1·0 − 1·0	+ 2·0 + 0	− 11·0 − 14·2	− 5·5 − 8·7	− 2·9 − 6·1	− 1·4 − 4·6	− 0 − 3·2	+ 1·6 − 1·6	+ 3·2 + 0
0·316– 0·4	− 1·7 − 4·2	− 0 − 2·5	+ 1·3 − 1·2	+ 2·5 + 0	− 11·0 − 14·9	− 6·0 − 9·9	− 3·2 − 7·1	− 1·7 − 5·6	− 0 − 3·9	+ 2·0 − 1·9	+ 3·9 + 0
0·401– 0·5	− 1·7 − 4·2	− 0 − 2·5	+ 1·3 − 1·2	+ 2·5 + 0	− 11·0 − 14·9	− 6·0 − 9·9	− 3·2 − 7·1	− 1·7 − 5·6	− 0 − 3·9	+ 2·0 − 1·9	+ 3·9 + 0
0·501– 0·63	− 2·1 − 5·0	− 0 − 2·9	+ 1·5 − 1·4	+ 2·9 + 0	− 11·0 − 15·5	− 6·0 − 10·5	− 3·6 − 8·1	− 2·1 − 6·6	− 0 − 4·5	+ 2·3 − 2·2	+ 4·5 + 0
0·631– 0·8	− 2·1 − 5·0	− 0 − 2·9	+ 1·5 − 1·4	+ 2·9 + 0	− 11·0 − 15·5	− 6·0 − 10·5	− 3·6 − 8·1	− 2·1 − 6·6	− 0 − 4·5	+ 2·3 − 2·2	+ 4·5 + 0
0·801– 1·0	− 2·6 − 6·0	− 0 − 3·4	+ 1·7 − 1·7	+ 3·4 + 0	− 12·0 − 17·5	− 6·5 − 12·0	− 3·9 − 9·4	− 2·6 − 8·1	− 0 − 5·5	+ 2·8 − 2·7	+ 5·5 + 0
1·001– 1·25	− 2·6 − 6·0	− 0 − 3·4	+ 1·7 − 1·7	+ 3·4 + 0	− 12·0 − 17·5	− 6·5 − 12·0	− 3·9 − 9·4	− 2·6 − 8·1	− 0 − 5·5	+ 2·8 − 2·7	+ 5·5 + 0
1·251– 1·6	− 3·2 − 7·2	− 0 − 4·0	+ 2·0 − 2·0	+ 4·0 + 0	− 12·0 − 18·0	− 6·5 − 12·5	− 4·2 − 10·2	− 3·2 − 9·2	− 0 − 6·0	+ 3·0 − 3·0	+ 6·0 + 0
1·601– 2·0	− 3·2 − 7·2	− 0 − 4·0	+ 2·0 − 2·0	+ 4·0 + 0	− 13·0 − 19·0	− 7·0 − 13·0	− 4·4 − 10·4	− 3·2 − 9·2	− 0 − 6·0	+ 3·0 − 3·0	+ 6·0 + 0
2·001– 2·5	− 3·9 − 8·6	− 0 − 4·7	+ 2·4 − 2·3	+ 4·7 + 0	− 13·0 − 20·5	− 7·5 − 15·0	− 5·5 − 13·0	− 3·9 − 11·4	− 0 − 7·5	+ 3·8 − 3·7	+ 7·5 + 0
2·501– 3·15	− 3·9 − 8·6	− 0 − 4·7	+ 2·4 − 2·3	+ 4·7 + 0	− 14·0 − 21·5	− 8·0 − 15·5	− 6·0 − 13·5	− 3·9 − 11·4	− 0 − 7·5	+ 3·8 − 3·7	+ 7·5 + 0
3·151– 4·0	− 4·8 − 10·3	− 0 − 5·5	+ 2·8 − 2·7	+ 5·5 + 0	− 15·0 − 23·5	− 8·5 − 17·0	− 6·5 − 15·0	− 4·8 − 13·3	− 0 − 8·5	+ 4·3 − 4·2	+ 8·5 + 0
4·001– 5·0	− 4·8 − 10·3	− 0 − 5·5	+ 2·8 − 2·7	+ 5·5 + 0	− 16·0 − 24·5	− 9·5 − 18·0	− 7·5 − 16·0	− 4·8 − 13·3	− 0 − 8·5	+ 4·3 − 4·2	+ 8·5 + 0
5·001– 5·6	− 5·5 − 12·0	− 0 − 6·5	+ 3·3 − 3·2	+ 6·5 + 0	− 19·0 − 29·0	− 9·5 − 19·5	− 8·0 − 18·0	− 5·5 − 15·5	− 0 − 10·0	+ 5·0 − 5·0	+ 10·0 + 0
5·601– 6·3	− 5·5 − 12·0	− 0 − 6·5	+ 3·3 − 3·2	+ 6·5 + 0	− 21·0 − 31·0	− 11·0 − 21·0	− 8·5 − 18·5	− 5·5 − 15·5	− 0 − 10·0	+ 5·0 − 5·0	+ 10·0 + 0
6·301– 7·1	− 6·0 − 13·0	− 0 − 7·0	+ 3·5 − 3·5	+ 7·0 + 0	− 23·0 − 34·0	− 12·0 − 23·0	− 9·0 − 20·0	− 6·0 − 17·0	− 0 − 11·0	+ 5·5 − 5·5	+ 11·0 + 0
7·101– 8·0	− 6·0 − 13·0	− 0 − 7·0	+ 3·5 − 3·5	+ 7·0 + 0	− 26·0 − 37·0	− 14·0 − 25·0	− 10·0 − 21·0	− 6·0 − 17·0	− 0 − 11·0	+ 5·5 − 5·5	+ 11·0 + 0
8·001– 9·0	− 7·0 − 14·5	− 0 − 7·5	+ 3·8 − 3·7	+ 7·5 + 0	− 30·0 − 42·0	− 15·0 − 27·0	− 11·0 − 23·0	− 7·0 − 19·0	− 0 − 12·0	+ 6·0 − 6·0	+ 12·0 + 0
9·001–10·0	− 7·0 − 14·5	− 0 − 7·5	+ 3·8 − 3·7	+ 7·5 + 0	− 33·0 − 45·0	− 17·0 − 29·0	− 11·0 − 23·0	− 7·0 − 19·0	− 0 − 12·0	+ 6·0 − 6·0	+ 12·0 + 0
10·001–11·2	− 7·5 − 15·5	− 0 − 8·0	+ 4·0 − 4·0	+ 8·0 + 0	− 37·0 − 50·0	− 19·0 − 32·0	− 12·0 − 25·0	− 7·5 − 20·5	− 0 − 13·0	+ 6·5 − 6·5	+ 13·0 + 0
11·201–12·5	− 7·5 − 15·5	− 0 − 8·0	+ 4·0 − 4·0	+ 8·0 + 0	− 41·0 − 54·0	− 21·0 − 34·0	− 13·0 − 26·0	− 7·5 − 20·5	− 0 − 13·0	+ 6·5 − 6·5	+ 13·0 + 0
12·501–14·0	− 8·5 − 17·5	− 0 − 9·0	+ 4·5 − 4·5	+ 9·0 + 0	− 46·0 − 60·0	− 24·0 − 38·0	− 14·0 − 28·0	− 8·5 − 22·5	− 0 − 14·0	+ 7·0 − 7·0	+ 14·0 + 0
14·001–16·0	− 8·5 − 17·5	− 0 − 9·0	+ 4·5 − 4·5	+ 9·0 + 0	− 52·0 − 66·0	− 27·0 − 41·0	− 16·0 − 30·0	− 8·5 − 22·5	− 0 − 14·0	+ 7·0 − 7·0	+ 14·0 + 0
16·001–18·0	− 9·5 − 19·5	− 0 − 10·0	+ 5·5 − 5·5	+ 10·0 + 0	− 59·0 − 74·0	− 31·0 − 46·0	− 17·0 − 32·0	− 9·5 − 24·5	− 0 − 15·0	+ 7·5 − 7·5	+ 15·0 + 0
18·001–20·0	− 9·5 − 19·5	− 0 − 10·0	+ 5·5 − 5·5	+ 10·0 + 0	− 66·0 − 81·0	− 34·0 − 49·0	− 19·0 − 34·0	− 9·5 − 24·5	− 0 − 15·0	+ 7·5 − 7·5	+ 15·0 + 0

Tolerance unit = 0·001 in.

7. Rounding-off Methods used in the Translation

To avoid the use of excessively precise tolerance dimensions, the theoretical values of all the fundamental tolerances and fundamental limits have been rounded off according to definite rules given below. In general, $x5$ is rounded to $x0$ if x is even, and to $(x + 1)0$ if x is odd (in accordance with A.S.A. practice).

(*a*) *Fundamental Tolerances*

Up to 0·0005 in.: tolerances are rounded off to the nearest 0·00005 in.

Above 0·0005 in. up to 0·005 in.: rounded to nearest 0·0001 in.

Above 0·005 in. up to 0·010 in.: rounded to nearest 0·0005 in.

Above 0·010 in.: rounded to nearest 0·001 in.

(*b*) *Fundamental Limits or Basic Deviations*

As for fundamental tolerances, except that limits even below 0·0005 in. are rounded off to the nearest 0·0001 in., since the effect on the resultant fit will be negligible. In the case of holes N to Z, 0·0005 in. has been added to certain figures to avoid two large limits each with a fourth decimal place.

(*c*) *Special Note regarding K, M, and N Holes*

The upper limits of the holes K6, K7, K8; M6, M8; N6, N7, and N8 have in certain cases been rounded off from the sum of the *un*rounded fundamental tolerances and fundamental limits (instead of the rounded) as otherwise the progression becomes uneven and illogical.

8. Special Notes on Tables of Fits

(*a*) *Shaft Limits*

(i) The shafts j8, j9, j10, j11, k8, k9, k10, and k11 are not settled for fits.

(ii) In the range 0·04 in. to 0·315 in. (1–10 mm) limits for k5, k6, k7, and m7 do not exist; the limits for j5, j6, j7, and n7 are to be used instead.

(iii) In the range 0·04 in. to 1·0 in. (1–24 mm) limits for t5, t6, and t7 do not exist; the limits for u5, u6, and u7 are to be used instead.

(iv) In the range 0·04 in. to 0·5 in. (1–14 mm) limits for v5, v6, and v7 do not exist; the limits for x5, x6, and x7 are to be used instead.

(v) In the range 0·04 in. to 0·63 in. (1–18 mm) limits for y6 and y7 do not exist; the limits for z6 and z7 are to be used instead.

(vi) The shafts v, x, y, and z are not to be considered as definite recommendations, but for trial only. The shafts v and y are to be avoided as much as possible.

(*b*) *Hole Limits*

(i) F9 is envisaged for precision work only up to a diameter of 1·25 in. (30 mm).

(ii) The holes J9, J10, J11, N9, N10, and N11 are not settled for fits.

(iii) In the range 0·04 in. to 0·315 in. (1–10 mm) limits for K6, K7, K8, and M8 do not exist; the limits for J6, J7, J8, and N8 are to be used instead.

(iv) In the range 0·04 in. to 1·0 in. (1–24 mm) limits for T6 and T7 do not exist; the limits for U6 and U7 are to be used instead.

(v) In the range 0·04 in. to 0·5 in. (1–14 mm) limits for V6 and V7 do not exist; the limits for X6 and X7 are to be used instead.

(vi) In the range 0·04 in. to 0·63 in. (1–18 mm) limits for Y7 do not exist; the limits for Z7 are to be used instead.

(vii) The holes V, X, Y, and Z are not to be considered as definite recommendations, but for trial only. The holes V and Y are to be avoided as much as possible.

9. Description of Fits

The description of fits given in the system is approximate only—

Fit	Description
H–h	Slide
H–j	Push
H–k	"Wringing"
H–m	Drive
H–n	Tight

10. Gauge Limits

A most comprehensive system of manufacturing, setting and wear limits for the gauges to be used with the I.S.A. System is published, and the information is reproduced, explained, and translated in Chapter VIII.

THE D.I.N. TOLERANCE SYSTEM

Information on the German D.I.N. Metric Tolerance System is given for reference, although it is unlikely to be used in this country in preference to the I.S.A. System, which is a later development and supersedes it.

The D.I.N. System is built around a fundamental tolerance unit $= 0{\cdot}005\sqrt[3]{D}$, and all basic deviations and tolerances are multiples of this unit. Both basic hole and basic shaft systems are tabled, but the unilateral basic hole system is preferred. Four grades of workmanship or tolerance are available (Superfine, Fine, Medium, and Coarse), but being an early system there is but one shaft associated with each hole.

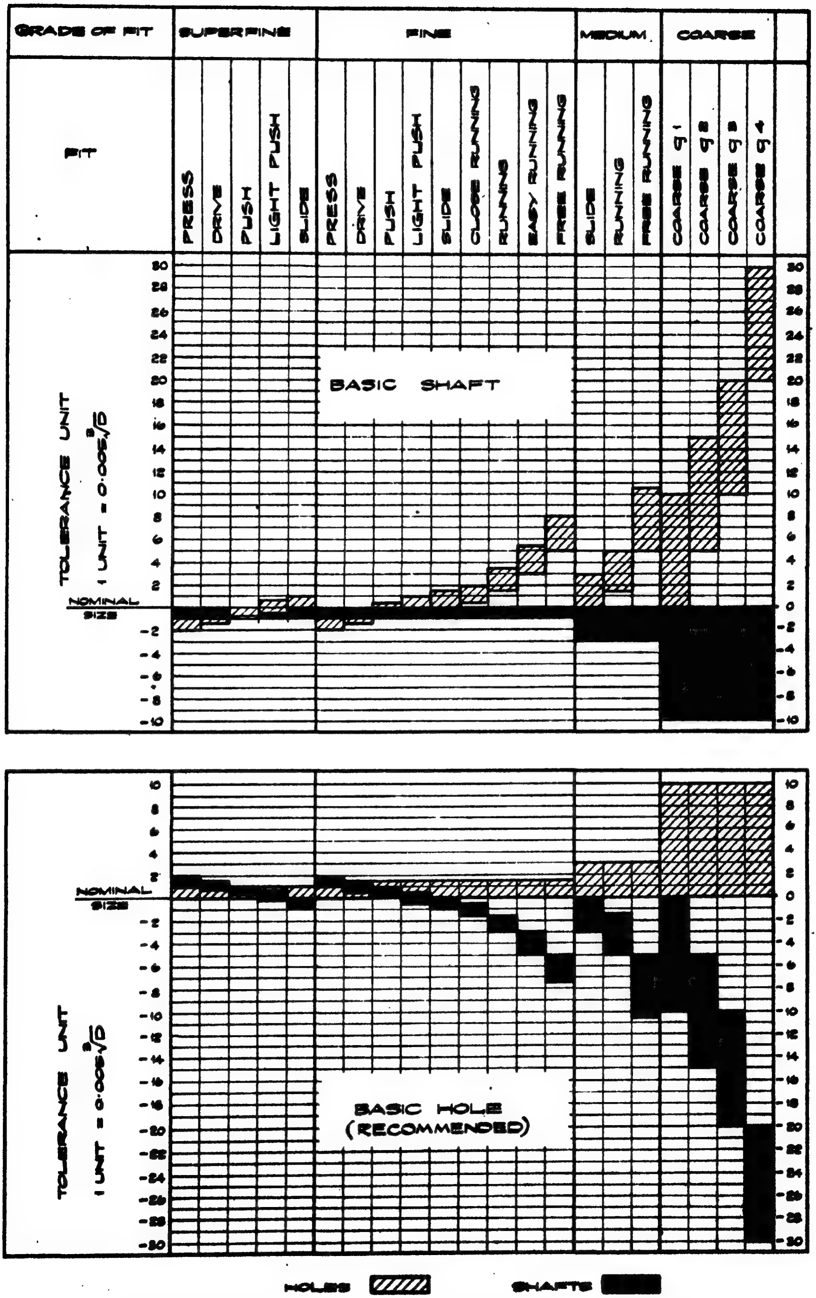


FIG. 73. D.I.N. LIMITS REPRESENTED DIAGRAMMATICALLY

The standard reference temperature is 20° C. (68° F.).

Table 25 lists the descriptions and code symbols of the various fits.

Fig. 73 shows the D.I.N. fits diagrammatically.

Fig. 74 gives the relation and comparison of the I.S.A. and D.I.N. systems.

Table 26 gives the full list of D.I.N. Tolerances.

Appendix 6 gives a translation of various technical German terms. It will be seen that the German letter symbols are derived from the first letter of the fit description.

TABLE 25

	Basic Hole		Basic Shaft	
	Hole	Shaft	Hole	Shaft
1. *Superfine Fits.*				
Press	eF			eF
Drive	eT			eT
Push	eH	eW	EB	eH
Light Push	eS			eS
Slide	eG			eG
2. *Fine Fits.*				
Heavy Press	P			P
Press	F			F
Drive	T			T
Push	H			H
Light Push	S	W	B	S
Slide	G			G
Close Running	EL			EL
Running	L			L
Easy Running	LL			LL
Free Running	WL			WL
3. *Medium Fits.*				
Slide	sG			sG
Running	sL	sW	sB	sL
Free	sLW			sW
4. *Coarse Fits.*				
g1	g1			g1
g2	g2	gW	gB	g2
g3	g3			g3
g4	g4			g4

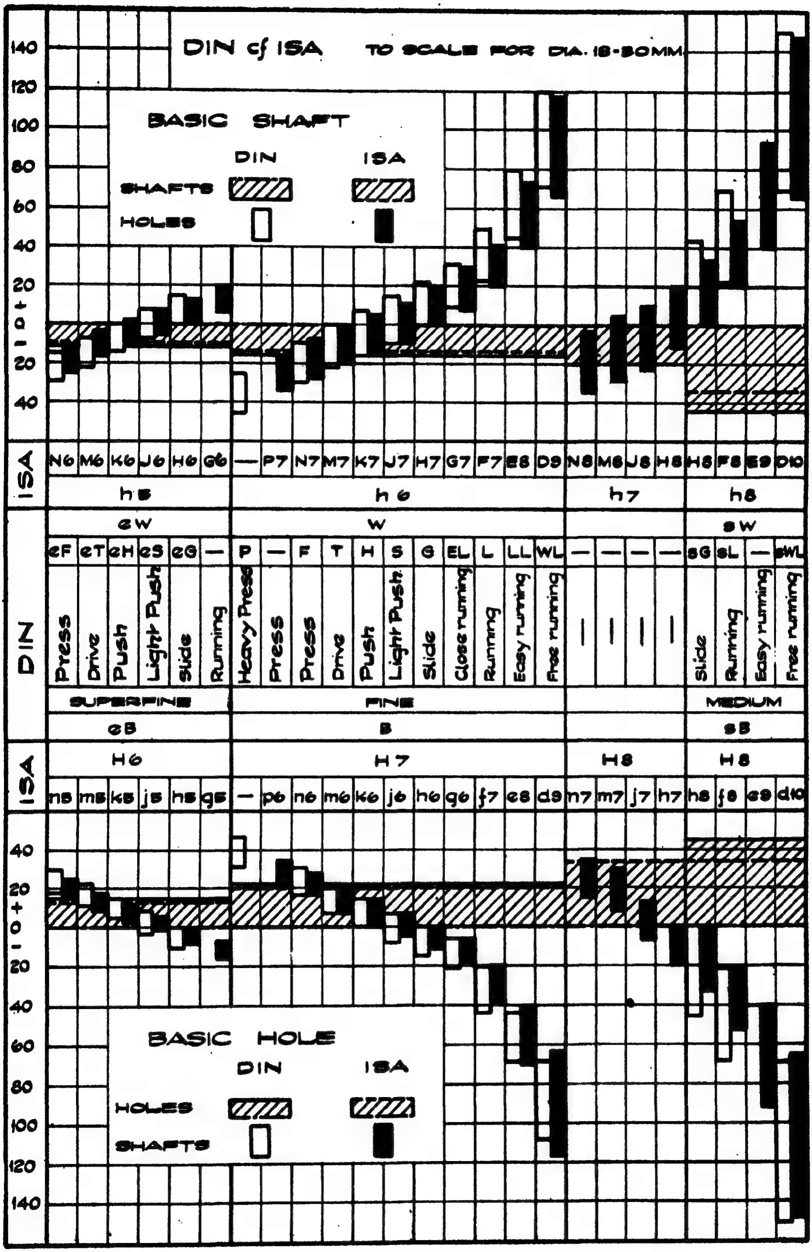


FIG. 74. THE RELATIONSHIP BETWEEN D.I.N. AND I.S.A. LIMITS

TABLE 26

	Hole	Shaft					Hole	Shaft				
	eB	eF	eT	eH	eS	eG	B	P	F	T	H	S
1 – 3	—	—	—	—	—	—	+ 9	+ 15	+ 12	+ 9	+ 6	+ 3
	—	—	—	—	—	—	+ 0	+ 10	+ 6	+ 3	+ 0	− 3
3·01– 6	+ 8	+ 15	+ 12	+ 8	+ 4	− 0	+ 12	+ 22	+ 15	+ 12	+ 8	+ 4
	+ 0	+ 10	+ 6	+ 2	− 2	− 6	+ 0	+ 15	+ 8	+ 4	+ 0	− 4
6·01– 10	+ 10	+ 20	+ 15	+ 10	+ 5	− 0	+ 15	+ 30	+ 20	+ 15	+ 10	+ 5
	+ 0	+ 12	+ 7	+ 2	− 2	− 7	+ 0	+ 20	+ 10	+ 5	+ 0	− 5
10·01– 18	+ 12	+ 25	+ 18	+ 12	+ 6	− 0	+ 18	+ 38	+ 25	+ 18	+ 12	+ 6
	+ 0	+ 15	+ 9	+ 3	− 3	− 9	+ 0	+ 25	+ 12	+ 6	+ 0	− 6
18·01– 30	+ 15	+ 30	+ 22	+ 15	+ 8	− 0	+ 22	+ 45	+ 30	+ 22	+ 15	+ 8
	+ 0	+ 18	+ 11	+ 4	− 4	− 11	+ 0	+ 32	+ 15	+ 8	+ 0	− 8
30·01– 50	+ 18	+ 35	+ 25	+ 18	+ 9	− 0	+ 25	+ 60	+ 35	+ 25	+ 18	+ 9
	+ 0	+ 22	+ 13	+ 4	− 4	− 13	+ 0	+ 40	+ 18	+ 9	+ 0	− 9
50·01– 80	+ 20	+ 40	+ 30	+ 20	+ 10	− 0	+ 30	+ 75	+ 40	+ 30	+ 20	+ 10
	+ 0	+ 25	+ 15	+ 5	− 5	− 15	+ 0	+ 55	+ 20	+ 10	+ 0	− 10
80·01–120	+ 22	+ 45	+ 35	+ 22	+ 11	− 0	+ 35	+ 90	+ 45	+ 35	+ 22	+ 11
	+ 0	+ 28	+ 17	+ 6	− 6	− 17	+ 0	+ 65	+ 22	+ 11	+ 0	− 11
120·01–180	+ 25	+ 50	+ 40	+ 25	+ 13	− 0	+ 40	+ 105	+ 50	+ 40	+ 25	+ 13
	+ 0	+ 32	+ 20	+ 7	− 7	− 20	+ 0	+ 80	+ 25	+ 13	+ 0	− 13
180·01–260	+ 30	+ 60	+ 45	+ 30	+ 15	− 0	+ 45	+ 130	+ 60	+ 45	+ 30	+ 15
	+ 0	+ 38	+ 22	+ 8	− 8	− 22	+ 0	+ 100	+ 30	+ 15	+ 0	− 15
260·01–360	+ 35	+ 70	+ 50	+ 35	+ 18	− 0	+ 50	+ 155	+ 70	+ 50	+ 35	+ 18
	+ 0	+ 43	+ 25	+ 9	− 9	− 25	+ 0	+ 120	+ 35	+ 18	+ 0	− 18
360·01–500	+ 40	+ 80	+ 60	+ 40	+ 20	− 0	+ 60	+ 180	+ 80	+ 60	+ 40	+ 20
	+ 0	+ 50	+ 28	+ 10	− 10	− 28	+ 0	+ 140	+ 40	+ 20	+ 0	− 20

TABLE 26 (*contd.*)

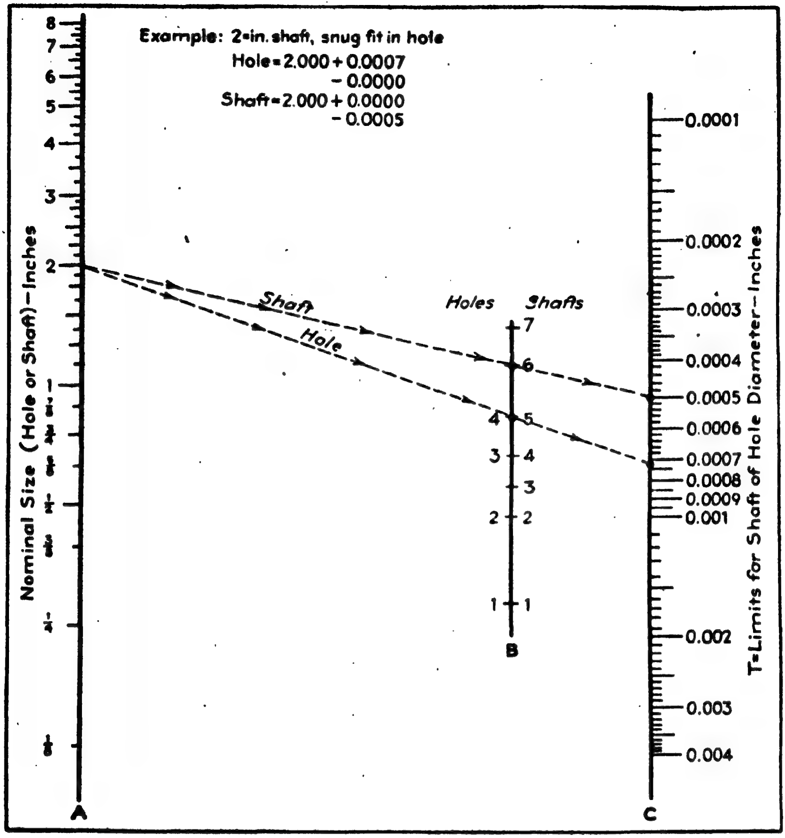
	Shaft	Hole					Shaft	Hole				
	eW	eF	eT	eH	eS	eG	W	P	F	T	H	S
1 – 3	—	—	—	—	—	—	− 0	− 7	− 3	− 0	+ 3	+ 6
	—	—	—	—	—	—	− 6	− 15	− 12	− 9	− 6	− 3
3·01– 6	− 0	− 8	− 4	− 0	+ 4	+ 8	− 0	− 10	− 4	− 0	+ 4	+ 8
	− 6	− 15	− 12	− 8	− 4	+ 0	− 8	− 22	− 15	− 12	− 8	− 4
6·01– 10	− 0	− 10	− 5	− 0	+ 5	+ 10	− 0	− 15	− 5	− 0	+ 5	+ 10
	− 7	− 20	− 15	− 10	− 5	+ 0	− 10	− 30	− 20	− 15	− 10	− 5
10·01– 18	− 0	− 12	− 6	− 0	+ 6	+ 12	− 0	− 20	− 6	− 0	+ 6	+ 12
	− 9	− 25	− 18	− 12	− 6	+ 0	− 12	− 38	− 25	− 18	− 12	− 6
18·01– 30	− 0	− 15	− 8	− 0	+ 8	+ 15	− 0	− 25	− 8	− 0	+ 8	+ 15
	− 11	− 30	− 22	− 15	− 8	+ 0	− 15	− 45	− 30	− 22	− 15	− 8
30·01– 50	− 0	− 18	− 9	− 0	+ 9	+ 18	− 0	− 35	− 9	− 0	+ 9	+ 18
	− 13	− 65	− 25	− 18	− 9	+ 0	− 18	− 60	− 35	− 25	− 18	− 9
50·01– 80	− 0	− 20	− 10	− 0	+ 10	+ 20	− 0	− 45	− 10	− 0	+ 10	+ 20
	− 15	− 40	− 30	− 20	− 10	+ 0	− 20	− 75	− 40	− 30	− 20	− 10
80·01–120	− 0	− 22	− 11	− 0	+ 11	+ 22	− 0	− 55	− 11	− 0	+ 11	+ 22
	− 17	− 45	− 35	− 22	− 11	+ 0	− 22	− 90	− 45	− 35	− 22	− 11
120·01–180	− 0	− 25	− 13	− 0	+ 13	+ 25	− 0	− 65	− 13	− 0	+ 13	+ 25
	− 20	− 50	− 40	− 25	− 13	+ 0	− 25	− 105	− 50	− 40	− 25	− 13
180·01–260	− 0	− 30	− 15	− 0	+ 15	+ 30	− 0	− 85	− 15	− 0	+ 15	+ 30
	− 22	− 60	− 45	− 30	− 15	+ 0	− 30	− 130	− 60	− 45	− 30	− 15
260·01–360	− 0	− 35	− 18	− 0	+ 18	+ 35	− 0	− 105	− 18	− 0	+ 18	+ 35
	− 25	− 70	− 50	− 35	− 18	+ 0	− 35	− 155	− 70	− 50	− 35	− 18
360·01–500	− 0	− 40	− 20	− 0	+ 20	+ 40	− 0	− 120	− 20	− 0	+ 20	+ 40
	− 28	− 80	− 60	− 40	− 20	+ 0	− 40	− 180	− 80	− 60	− 40	− 20

German D.I.N. Limits *Hole Basis (Unilateral Hole)*

					Hole	Shaft			Hole	Shaft			
G	EL	L	LL	WL	sB	sG	sL	sWL	gB	g1	g2	g3	g4
− 0	− 3	− 9	− 18	− 30	+ 18	− 0	− 9	− 30	+ 50	− 0	− 30	− 50	− 100
− 6	− 9	− 18	− 30	− 50	+ 0	− 18	− 30	− 60	+ 0	− 50	− 80	− 100	− 180
− 0	− 4	− 12	− 25	− 40	+ 25	− 0	− 12	− 40	+ 80	− 0	− 40	− 80	− 150
− 8	− 12	− 25	− 40	− 60	+ 0	− 25	− 40	− 80	+ 0	− 80	− 120	− 150	− 250
− 0	− 5	− 15	− 30	− 50	+ 30	− 0	− 15	− 50	+ 100	− 0	− 50	− 100	− 200
− 10	− 15	− 30	− 50	− 75	+ 0	− 30	− 50	− 100	+ 0	− 100	− 150	− 200	− 300
− 0	− 6	− 18	− 35	− 60	+ 35	− 0	− 18	− 60	+ 100	− 0	− 60	− 100	− 250
− 12	− 18	− 35	− 60	− 90	+ 0	− 35	− 60	− 120	+ 0	− 100	− 200	− 250	− 350
− 0	− 8	− 22	− 45	− 70	+ 45	− 0	− 22	− 70	+ 150	− 0	− 70	− 150	− 300
− 15	− 22	− 45	− 70	− 110	+ 0	− 45	− 70	− 150	+ 0	− 150	− 250	− 300	− 450
− 0	− 9	− 25	− 50	− 80	+ 50	− 0	− 25	− 80	+ 150	− 0	− 80	− 150	− 350
− 18	− 25	− 50	− 80	− 130	+ 0	− 50	− 80	− 180	+ 0	− 150	− 250	− 350	− 500
− 0	− 10	− 30	− 60	− 100	+ 60	− 0	− 30	− 100	+ 200	− 0	− 100	− 200	− 400
− 20	− 30	− 60	− 100	− 150	+ 0	− 60	− 100	− 200	+ 0	− 200	− 300	− 400	− 600
− 0	− 11	− 35	− 70	− 120	+ 70	− 0	− 35	− 120	+ 200	− 0	− 120	− 200	− 450
− 22	− 35	− 70	− 120	− 180	+ 0	− 70	− 120	− 250	+ 0	− 200	− 350	− 450	− 700
− 0	− 13	− 40	− 80	− 140	+ 80	− 0	− 40	− 140	+ 250	− 0	− 140	− 250	− 500
− 25	− 40	− 80	− 140	− 200	+ 0	80	− 140	− 280	+ 0	− 250	− 400	− 500	− 800
− 0	− 15	− 45	− 90	− 150	+ 90	− 0	− 45	− 150	+ 250	− 0	− 150	− 250	− 550
− 30	− 45	− 90	− 150	− 220	+ 0	− 90	− 150	− 320	+ 0	− 250	− 450	− 550	− 900
− 0	− 18	− 50	− 100	− 170	+ 100	− 0	− 50	− 170	+ 300	− 0	− 170	− 300	− 600
− 35	− 50	− 100	− 170	− 250	+ 0	− 100	− 170	− 350	+ 0	− 300	− 500	− 600	− 1000
− 0	− 20	− 60	− 120	− 200	+ 120	− 0	− 60	− 200	+ 350	− 0	− 200	− 350	− 700
− 40	− 60	− 120	− 200	− 280	+ 0	− 120	− 200	− 400	+ 0	− 350	− 550	− 700	− 1100

German D.I.N. Limits *Shaft Basis (Unilateral Shaft)*

					Shaft	Hole			Shaft	Hole			
G	EL	L	LL	WL	sW	sG	sL	sWL	gW	g1	g2	g3	g4
+ 9	+ 12	+ 20	+ 35	+ 50	− 0	+ 18	+ 30	+ 60	− 0	+ 50	+ 80	+ 100	+ 180
+ 0	+ 3	+ 9	+ 18	+ 30	− 18	+ 0	+ 9	+ 30	− 50	+ 0	+ 30	+ 50	+ 100
+ 12	+ 15	+ 30	+ 45	+ 60	− 0	+ 25	+ 40	+ 80	− 0	+ 80	+ 120	+ 150	+ 250
+ 0	+ 4	+ 12	+ 25	+ 40	− 25	+ 0	+ 12	+ 40	− 80	+ 0	+ 40	+ 80	+ 150
+ 15	+ 20	− 35	+ 55	+ 80	− 0	+ 30	+ 50	+ 100	− 0	+ 100	+ 150	+ 200	+ 300
+ 0	+ 5	+ 15	+ 30	+ 50	− 30	+ 0	+ 15	+ 50	− 100	+ 0	+ 50	+ 100	+ 200
+ 18	+ 25	+ 40	+ 65	+ 100	− 0	+ 35	+ 60	+ 120	− 0	+ 100	+ 200	+ 250	+ 350
+ 0	+ 6	+ 18	+ 35	+ 60	− 35	+ 0	+ 18	+ 60	− 100	+ 0	+ 60	+ 100	+ 250
+ 22	+ 30	+ 50	+ 80	+ 120	− 0	+ 45	+ 70	+ 150	− 0	+ 150	+ 250	+ 300	+ 450
+ 0	+ 8	+ 22	+ 45	+ 70	− 45	+ 0	+ 22	+ 70	− 150	+ 0	+ 70	+ 150	+ 300
+ 25	+ 35	+ 60	+ 95	+ 140	− 0	+ 50	+ 80	+ 180	− 0	+ 150	+ 250	+ 350	+ 500
+ 0	+ 9	+ 25	+ 50	+ 80	− 50	+ 0	+ 25	+ 80	− 150	+ 0	+ 80	+ 150	+ 350
+ 30	+ 40	+ 70	+ 110	+ 160	− 0	+ 60	+ 100	+ 200	− 0	+ 200	+ 300	+ 400	+ 600
+ 0	+ 10	+ 30	+ 60	+ 100	− 60	+ 0	+ 30	+ 100	− 200	+ 0	+ 100	+ 200	+ 400
+ 35	+ 45	+ 80	+ 130	+ 180	− 0	+ 70	+ 120	+ 250	− 0	+ 200	+ 350	+ 450	+ 700
+ 0	+ 11	+ 35	+ 70	+ 120	− 70	+ 0	+ 35	+ 120	− 200	+ 0	+ 120	+ 200	+ 450
+ 40	+ 50	+ 95	+ 150	+ 210	− 0	+ 80	+ 140	+ 280	− 0	+ 250	+ 400	+ 500	+ 800
+ 0	+ 13	+ 40	+ 80	+ 140	− 80	+ 0	+ 40	+ 140	− 250	+ 0	+ 140	+ 250	+ 500
+ 45	+ 60	+ 105	+ 170	+ 240	− 0	+ 90	+ 150	+ 320	− 0	+ 250	+ 450	+ 550	+ 900
+ 0	+ 15	+ 45	+ 90	+ 150	− 90	+ 0	+ 45	+ 150	− 250	+ 0	+ 150	+ 250	+ 550
+ 50	+ 70	+ 120	+ 190	+ 270	− 0	+ 100	+ 170	+ 350	− 0	+ 300	+ 500	+ 600	+ 1000
+ 0	+ 18	+ 50	+ 100	+ 170	− 100	+ 0	+ 50	+ 170	− 300	+ 0	+ 170	+ 300	+ 600
+ 60	+ 80	+ 140	+ 220	+ 300	− 0	+ 120	+ 200	+ 400	− 0	+ 350	+ 550	+ 700	+ 1100
+ 0	+ 20	+ 60	+ 120	+ 200	− 120	+ 0	+ 60	+ 200	− 350	+ 0	+ 200	+ 350	+ 700



Standards for Metal Fits		Swing Point—Line B		Diametral Limits	
Class of Fit	Method of Assembly	Hole	Shaft	Hole	Shaft
1. Loose	Interchangeable	(1)	(1)	+ T, − 0	− T, − 2T
2. Free	"	(2)	(2)	+ T, − 0	− T, − 2T
3. Medium	"	(3)	(4)	+ T, − 0	− T, − 2T
4. Snug	"	(4)	(6)	+ T, − 0	+ 0, − T
5. Wringing	Selective	(4)	(6)	+ T, − 0	+ T, − 0
6. Tight	"	(4)	(7)	+ T, − 0	+ T, + 3T
7. Medium Force	"	(4)	(5)	+ T, − 0	+ T, + 1[illegible]T
8. Heavy Force or Shrink		(4) (3)[1]	(5)[2]	+ T, − 0	+ T, + 1[illegible]T

[1] Steel only. [2] Cast iron.

Fig. 75. "The Machinist" Chart of Allowances for Shaft Fits

To use chart—

1. Find size of hole or shaft on line A.
2. Find swing point on line B for proper fit. Note hole fits are on the left-hand side, shaft fits on the right-hand side.
3. Lay straight edge across these two points to locate point on line C.
4. Determine diametral limits from reading on line C and values given in above table.

MISCELLANEOUS TOLERANCE SYSTEMS

Under this heading come one or two systems evolved by individuals or particular firms, usually toolmakers, for special reasons, or often to assist users of their tools. These systems have little interest for future use, but are included for the sake of completeness and because they may still be in use. Ball-bearing makers' limits are dealt with separately in the next chapter.

1. The Brown and Sharpe Table of "Allowances for Fits" (*Vide B. & S. Handbook*)

Table 27 gives the range of fit deviations or allowances recommended by this company and which therefore have to be divided up as tolerances on hole and shaft. A normal and suggested division would be two-thirds on the hole and one-third on the shaft.

TABLE 27

Description of Fit	Diameter—inches				
	0–0·5	0·501–1	1·001–2	2·001–3·5	3·501–6
Running fit—up to 600 r.p.m. Light duty	− 0·5 − 1	− 0·75 − 1·5	− 1·5 − 2·5	− 2 − 3	− 2·5 − 4
Running fit—over 600 r.p.m. Heavy duty	− 0·5 − 1	− 1 − 2	− 2 − 3	− 3 − 4	− 4 − 5
Sliding fit	− 0·5 − 1	− 0·75 − 1·5	− 1·5 − 2·5	− 2 − 3	− 2·5 − 4
Key fit—clamp endwise .	− 0 − 0·25	− 0 − 0·5	− 0 − 0·5	− 0 − 0·5	− 0 − 0·75
Assembly with least play .	+ 0·25 + 0	+ 0·5 + 0	+ 0·5 + 0	+ 0·5 + 0	+ 0·75 + 0
Drive fit—easy to drive .	+ 0·25 + 0	+ 0·5 + 0·25	+ 0·75 + 0·5	+ 1 + 0·5	+ 1 + 0·5
Drive fit—permanent assembly	+ 1 + 0·5	+ 1 + 0·5	+ 1 + 0·5	+ 1·25 + 0·75	+ 1·5 + 1
Force fit—permanent press fit	+ 1 + 0·75	+ 2 + 1	+ 3 + 2	+ 4 + 3	+ 5 + 4

Unit = 0·001 in.

2. "The Machinist" Chart of Allowances for Shaft Fits. (This is reproduced in Fig. 75.)

CHAPTER V

STANDARD LIMIT AND TOLERANCE SYSTEMS (*contd.*)

BALL AND ROLLER BEARING LIMITS AND TOLERANCES

THERE is no aspect of the tolerance problem more involved and confusing than arises from the use of ball and roller bearings. This is due in the main to—

(i) Complete lack of agreement between one bearing maker and another as to what should be done.

(ii) The consequent existence of several solutions to a given problem, each maker's catalogue being different.

(iii) The fact that Continental and British metric bearings are not made to the same tolerances.

(iv) The fact that no tolerance system except the I.S.A. takes bearing fits into account and thus all inch fits are special. (The S.K.F. Company has translated the I.S.A. system into inches, but owing to the difference in the limits on the bearings themselves the limits of their translated shafts and holes have had to be altered, and are not the same as those in a true translation such as in Chapter IV.)

In the following paragraphs, some attempt has been made to sort out the confusion, but it is to be hoped that in due course the bearing makers will do this themselves. The reader is recommended to read *Rolling Bearings*, by R. K. Allan.

1. Manufacturing Limits for Ball and Roller Bearings

In order to visualize the significance of the tolerances on bearing diameters, it is first necessary to consider the possible errors.

Both the outside and inside diameter may be out of round, oval, etc., as measured by rolling the bearing between a flat plate and a dial indicator (Fig. 76). The mean diameter (D_m, d_m) is

$$= \frac{D_{max} + D_{min}}{2} \text{ or } \frac{d_{max} + d_{min}}{2}$$

Since errors of ovality are less important than errors of mean diameter, the bearing limits should take this into account.

The inner and outer diameters may rotate eccentrically with respect to each other.

With the inner diameter fitted to a mandrel and the outer diameter under a dial gauge, these errors can be measured (Fig. 77).

Eccentricity of the *inner* race is checked by rotating the mandrel keeping the outer race fixed (position *b*).

Eccentricity of the *outer* race is checked by rotating the outer ring, keeping the inner race fixed (position *b*).

The inner and outer races may rotate with wobble with respect to each other. This can be verified with the same set-up as for eccentricity, using side face indicators. Wobble of the *inner* race is checked by rotating the inner race, keeping the outer race fixed and recording off the side of the *outer* race (position *c*).

Wobble of the *outer* race is checked by rotating the outer race,

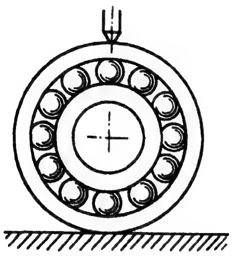


FIG. 76

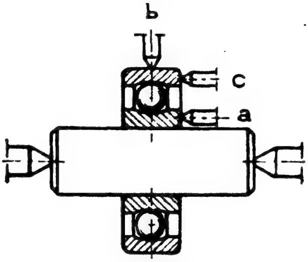


FIG. 77

keeping the inner race fixed, and recording off the side of the *outer* race (position *c*).

The inner race side faces may not be ground true with the inner bore diameter, and this is usually comparatively important in mounting, particularly with sliding shaft fits. This can be verified with the previous set-up by indicating on the side face while the race is rotated (position *d*). (Note that the error of inner race wobble includes this error.)

Inner race side face parallelism, more or less important, is specified as measured in any usual way. Outer race parallelism is not so important and is not specified.

The whole of the above process is that adopted for I.S.A. bearings and covers all errors (Table 28). The British method as contained in B.S. 292—1927 is a simplification (Table 29). This standard

(i) ignores the difference in importance between errors of mean and extreme diameters;

(ii) does not specify eccentricity errors for inner and outer races separately but uses the same value;

(iii) does not relate tolerances to the type of bearing (light, heavy, etc.);

(iv) does not distinguish between inner race wobble and side-face out of truth;

(v) does specify inner and outer race side parallelism.

American bearing tolerances (Classes I–IV) are given in Tables 30–33. The Class I tolerances are based on a range of bearings and tolerances standardized by the S.A.E., the bearing nominal dimensions being the same as the I.S.A. series. The limits, however, are different, although admittedly not by much. For some peculiar reason, although the bearings are metric, the limits on the bearings themselves and on the housings and shafts required are quoted in inches. Tables of official conversions giving the bearing dimensions in inches are published to avoid inevitable errors in continual conversion.

TABLE 28

I.S.A. BALL AND ROLLER BEARING TOLERANCES

1. TOLERANCES ON OUTER RACE

Nominal Outside Dia. D mm	Permissible Limits (unit = 0·001 mm): On Mean Dia. (D mean)		On Diameter D: Light Series		Normal Series		Heavy Series		On Width	Outer Race Eccentricity, max.	Outer Race Wobble, max.
10·1– 18	0	− 8	+ 1	− 9					See below*	15	40
18·1– 30	0	− 9	+ 2	− 11						15	40
30·1– 50	0	− 11	+ 3	− 14	+ 3	− 14				20	40
50·1– 80	0	− 13	+ 4	− 17	+ 4	− 17	+ 3	− 16		25	40
80·1–120	0	− 15	+ 6	− 21	+ 5	− 20	+ 4	− 19		35	45
120·1–150	0	− 18	+ 7	− 25	+ 6	− 24	+ 5	− 23		40	50
150·1–180	0	− 25	+ 8	− 33	+ 6	− 31	+ 5	− 30		45	60
180·1–250	0	− 30	+ 9	− 39	+ 7	− 37	+ 6	− 36		50	70
250·1–315	0	− 35	+ 10	− 45	+ 8	− 43	+ 7	− 42		60	80
315·1–400	0	− 40	+ 11	− 51	+ 9	− 49	+ 8	− 48		70	90
400·1–500	0	− 45	+ 13	− 58	+ 11	− 56	+ 9	− 54		80	100
500·1–630	0	− 50	+ 15	− 65	+ 12	− 62	+ 10	− 60		100	120

2. TOLERANCES ON INNER RACE

Nominal Inside Bore Dia. d mm	Permissible Limits (unit = 0·001 mm): On Mean Inner Dia. (d mean)		On Inner Dia. d		On Width: Ball and Parallel Roller Bearings		Tapered Roller Bearings		Error in Parallelism, max.	Side Face Truth Error, max.	Inner Race Eccentricity, max.	Inner Race Wobble, max.
− 30	0	− 10	+ 3	− 13	0	− 100	0	− 200	20	20	15	40
30·1– 50	0	− 12	+ 3	− 15	0	− 120	0	− 240	20	20	15	40
50·1– 80	0	− 15	+ 4	− 19	0	− 150	0	− 300	25	25	20	50
80·1–120	0	− 20	+ 5	− 25	0	− 200	0	− 400	25	25	25	50
120·1–180	0	− 25	+ 6	− 31	0	− 250	0	− 500	30	30	30	60
180·1–250	0	− 30	+ 8	− 38	0	− 300	0	− 600	30	30	40	60
250·1–315	0	− 35	+ 9	− 44	0	− 350	0	− 700	35	35	50	70
315·1–400	0	− 40	+ 10	− 50	0	− 400	0	− 800	40	40	60	80

* The width limits of the outer races of ball and parallel roller bearings are the same as for their inner races; no limits are specified for the outer race width of tapered roller bearings.

TABLE 29

B.S. 292 Ball and Roller Bearing Tolerances

A. Metric Bearings (Tolerance unit = 0·001 mm)

Nominal Diameter, Outside or Bore (D or d) mm	Journal Bearings: Limits: Outside Dia. D	Journal Bearings: Limits: Inner Bore d	Journal Bearings: Limits: Width, either Race	Journal Bearings: Error in Parallelism, either Race, max.	Journal Bearings: Error in Eccentricity, max.	Journal Bearings: Error in Wobble, max.	Thrust Bearings: Limits: Outside Dia. D	Thrust Bearings: Limits: Inner Bore d	Thrust Bearings: Limits: Width of Assembly	Thrust Bearings: Error in Eccentricity, max.	Thrust Bearings: Error in Wobble, max.
0 – 30	0, – 13	+ 5, – 10	0, – 51	13	10	25	0, – 25	+ 25, 0	± 50	20	10
30·1– 75	0, – 13	+ 5, – 10	0, – 51	13	10	25	0, – 25	+ 25, 0	± 50	30	15
75·1– 80	0, – 20	+ 5, – 10	0, – 51	13	10	25	0, – 25	+ 25, 0	± 50	30	20
80·1–100	0, – 20	+ 5, – 13	0, – 51	13	10	25	0, – 25	+ 25, 0	± 50	40	20
100·1–150	0, – 20	+ 5, – 13	0, – 51	13	15	38	0, – 25	+ 25, 0	± 50	40	25
150·1–180	0, – 25	+ 5, – 13	0, – 51	13	15	51	0, – 25	+ 25, 0	± 50	40	25
180·1–200	0, – 25	+ 5, – 13	0, – 51	13	15	51	0, – 25	+ 25, 0	± 50	50	25
200·1–350	0, – 25	—	0, – 51	13	25	51	0, – 25	+ 25, 0	± 50	50	25
350·1–360	0, – 25	—	0, – 51	13	38	51	0, – 25	+ 25, 0	± 50	50	25
360·1–420	0, – 25	—	0, – 51	13	38	51	0, – 25	+ 25, 0	± 50	80	25

B. Inch Bearings (Tolerance unit = 0·001 in.)

Nominal Diameter, Outside or Bore (D or d) mm	Journal Bearings: Limits: Outside Dia. D	Journal Bearings: Limits: Inner Bore d	Journal Bearings: Limits: Width, either Race	Journal Bearings: Error in Parallelism, either Race max.	Journal Bearings: Error in Eccentricity, max.	Journal Bearings: Error in Wobble, max.	Thrust Bearings: Limits: Outside Dia. D	Thrust Bearings: Limits: Inner Bore d	Thrust Bearings: Limits: Width of Assembly	Thrust Bearings: Error in Eccentricity, max.	Thrust Bearings: Error in Wobble, max.
0 – 1·125	– 0·3, – 0·8	+ 0·2, – 0·3	0, – 2	0·5	0·4	1	0, – 1	+ 1, 0	± 2	0·8	0·4
1·126– 1·99	– 0·3, – 0·8	+ 0·2, – 0·3	0, – 2	0·5	0·4	1	0, – 1	+ 1, 0	± 2	1·2	0·4*
2 – 2·99	– 0·5, – 1	+ 0·2, – 0·3	0, – 2	0·5	0·4	1	0, – 1	+ 1, 0	± 2	1·2*	0·6*
3 – 3·75	– 0·8, – 1·3	+ 0·2, – 0·3	0, – 2	0·5	0·4	1	0, – 1	+ 1, 0	± 2	1·6	0·8
3·76 – 4	– 0·8, – 1·3	+ 0·2, – 0·3	0, – 2	0·5	0·4	1·5	0, – 1	+ 1, 0	± 2	1·6	0·8
4·01 – 4·99	– 0·8, – 1·3	+ 0·2, – 0·3	0, – 2	0·5	0·6	1·5	0, – 1	+ 1, 0	± 2	1·6	1
5 – 6	– 1·3, – 1·8	+ 0·2, – 0·3	0, – 2	0·5	0·6	1·5	0, – 1	+ 1, 0	± 2	1·6	1
6·01 – 7	– 1·3, – 1·8	+ 0·2, – 0·5	0, – 2	0·5	0·6	2	0, – 1	+ 1, 0	± 2	1·6	1
7·01 – 8	– 1·3, – 1·8	+ 0·2, – 0·5	0, – 2	0·5	0·6	2	0, – 1	+ 1, 0	± 2	2	1
8·01 –11·99	– 1·3, – 1·8	+ 0·2, – 0·5	0, – 2	0·5	1·0	2	0, – 1	+ 1, 0	± 2	2	1
12 – 14	– 1·3, – 2·3	+ 0·2, – 0·5	0, – 2	0·5	1·0	2	0, – 1	+ 1, 0	± 2	2	1
14·01 –19·99	– 1·3, – 2·3	+ 0·2, – 0·5	0, – 2	0·5	1·[illegible]	2	0, – 1	+ 1, 0	± 2	2·4	1

* Includes 2 in. and 3 in.

TABLE 30

American "Annular Bearing Engineers' Committee" Ball and Roller Bearing Limits, Class I

(Tolerance unit = 0·001 in.)

Bearing Bore, mm	Outside Dia. — 0			Bearing Bore — 0	Eccentricity		Width — 0
	Light	Medium	Heavy		Inner Ring, max.	Outer Ring, max.	
10	— 0·4	— 0·5	—	— 0·3	0·4	0·8	— 5
12	— 0·5	— 0·5	—	— 0·3	0·4	0·8	— 5
15	— 0·5	— 0·5	—	— 0·3	0·4	0·8	— 5
17	— 0·5	— 0·5	— 0·6	— 0·3	0·4	0·8	— 5
20	— 0·5	— 0·6	— 0·6	— 0·4	0·4	0·8	— 5
25	— 0·6	— 0·6	— 0·6	— 0·4	0·4	0·8	— 5
30	— 0·6	— 0·6	— 0·8	— 0·4	0·8	1·2	— 5
35	— 0·6	— 0·6	— 0·8	— 0·5	0·8	1·2	— 5
40	— 0·6	— 0·8	— 0·8	— 0·5	0·8	1·2	— 5
45	— 0·8	— 0·8	— 0·8	— 0·5	1·0	1·6	— 5
50	— 0·8	— 0·8	— 1·0	— 0·5	1·0	1·6	— 5
55	— 0·8	— 0·8	— 1·0	— 0·6	1·0	1·6	— 5
60	— 0·8	— 1·0	— 1·0	— 0·6	1·0	1·6	— 5
65	— 0·8	— 1·0	— 1·0	— 0·6	1·0	1·6	— 5
70	— 1·0	— 1·0	— 1·0	— 0·6	1·0	1·6	— 5
75	— 1·0	— 1·0	— 1·2	— 0·6	1·0	1·6	— 5
80	— 1·0	— 1·0	— 1·2	— 0·6	1·2	1·8	— 5
85	— 1·0	— 1·0	— 1·2	— 0·8	1·2	1·8	— 5
90	— 1·0	— 1·2	— 1·2	— 0·8	1·2	1·8	— 5
95	— 1·0	— 1·2	— 1·2	— 0·8	1·2	1·8	— 5
100	— 1·0	— 1·2	— 1·2	— 0·8	1·2	1·8	— 5
105	— 1·2	— 1·2	— 1·2	— 0·8	1·2	1·8	— 5
110	— 1·2	— 1·2	— 1·6	— 0·8	1·2	1·8	— 5
120	— 1·2	— 1·2	— 1·6	— 0·8	1·4	2·0	— 5
130	— 1·2	— 1·6	— 1·6	— 1·0	1·4	2·0	— 5
140	— 1·2	— 1·6	— 1·6	— 1·0	1·4	2·0	— 5

TABLE 31

AMERICAN "ANNULAR BEARING ENGINEERS' COMMITTEE" BALL AND ROLLER BEARING LIMITS, CLASS II

(Tolerance unit = 0·001 in.)

Bearing Bore, mm	Outside Dia. — 0			Bearing Bore — 0	Eccentricity		Width, — 0
	Light	Medium	Heavy		Inner Ring, max.	Outer Ring, max.	
10	— 0·4	— 0·5	—	— 0·2	0·4	0·8	— 5
12	— 0·5	— 0·5	—	— 0·2	0·4	0·8	— 5
15	— 0·5	— 0·5	—	— 0·2	0·4	0·8	— 5
17	— 0·5	— 0·5	— 0·6	— 0·2	0·4	0·8	— 5
20	— 0·5	— 0·6	— 0·6	— 0·2	0·4	0·8	— 5
25	— 0·6	— 0·6	— 0·6	— 0·2	0·4	0·8	— 5
30	— 0·6	— 0·6	— 0·8	— 0·2	0·8	1·2	— 5
35	— 0·6	— 0·6	— 0·8	— 0·3	0·8	1·2	— 5
40	— 0·6	— 0·8	— 0·8	— 0·3	0·8	1·2	— 5
45	— 0·8	— 0·8	— 0·8	— 0·3	1·0	1·6	— 5
50	— 0·8	— 0·8	— 1·0	— 0·3	1·0	1·6	— 5
55	— 0·8	— 0·8	— 1·0	— 0·3	1·0	1·6	— 5
60	— 0·8	— 1·0	— 1·0	— 0·3	1·0	1·6	— 5
65	— 0·8	— 1·0	— 1·0	— 0·3	1·0	1·6	— 5
70	— 1·0	— 1·0	— 1·0	— 0·3	1·0	1·6	— 5
75	— 1·0	— 1·0	— 1·2	— 0·3	1·0	1·6	— 5
80	— 1·0	— 1·0	— 1·2	— 0·3	1·2	1·8	— 5
85	— 1·0	— 1·0	— 1·2	— 0·5	1·2	1·8	— 5
90	— 1·0	— 1·2	— 1·2	— 0·5	1·2	1·8	— 5
95	— 1·0	— 1·2	— 1·2	— 0·5	1·2	1·8	— 5
100	— 1·0	— 1·2	— 1·2	— 0·5	1·2	1·8	— 5
105	— 1·2	— 1·2	— 1·2	— 0·5	1·2	1·8	— 5
110	— 1·2	— 1·2	— 1·6	— 0·5	1·2	1·8	— 5
120	—	—	—	—	—	—	—
130	—	—	—	—	—	—	—
140	—	—	—	—	—	—	—

TABLE 32

AMERICAN "ANNULAR BEARING ENGINEERS' COMMITTEE"
BALL AND ROLLER BEARING LIMITS, CLASS III

(Tolerance unit = 0·001 in.)

Bearing Bore, mm	Outside Dia. – 0			Bearing Bore – 0	Eccentricity		Width – 0
	Light	Medium	Heavy		Inner Ring, max.	Outer Ring, max.	
10	– 0·3	– 0·4	—	– 0·2	0·2	0·8	– 5
12	– 0·4	– 0·4	—	– 0·2	0·2	0·8	– 5
15	– 0·4	– 0·4	—	– 0·2	0·2	0·8	– 5
17	– 0·4	– 0·4	– 0·4	– 0·2	0·2	0·8	– 5
20	– 0·4	– 0·4	– 0·4	– 0·2	0·2	0·8	– 5
25	– 0·4	– 0·4	– 0·4	– 0·2	0·2	0·8	– 5
30	– 0·4	– 0·4	– 0·5	– 0·2	0·4	1·2	– 5
35	– 0·4	– 0·4	– 0·5	– 0·3	0·4	1·2	– 5
40	– 0·4	– 0·5	– 0·5	– 0·3	0·4	1·2	– 5
45	– 0·5	– 0·5	– 0·5	– 0·3	0·5	1·6	– 5
50	– 0·5	– 0·5	– 0·7	– 0·3	0·5	1·6	– 5
55	– 0·5	– 0·5	– 0·7	– 0·3	0·5	1·6	– 5
60	– 0·5	– 0·7	– 0·7	– 0·3	0·5	1·6	– 5
65	– 0·5	– 0·7	– 0·7	– 0·3	0·5	1·6	– 5
70	– 0·7	– 0·7	– 0·7	– 0·3	0·5	1·6	– 5
75	– 0·7	– 0·7	– 0·9	– 0·3	0·5	1·6	– 5
80	– 0·7	– 0·7	– 0·9	– 0·3	0·6	1·8	– 5
85	– 0·7	– 0·7	– 0·9	– 0·5	0·6	1·8	– 5
90	– 0·7	– 0·9	– 0·9	– 0·5	0·6	1·8	– 5
95	– 0·7	– 0·9	– 0·9	– 0·5	0·6	1·8	– 5
100	– 0·7	– 0·9	– 0·9	– 0·5	0·6	1·8	– 5
105	– 0·9	– 0·9	– 0·9	– 0·5	0·6	1·8	– 5
110	– 0·9	– 0·9	– 1·2	– 0·5	0·6	1·8	– 5
120	—	—	—	—	—	—	—
130	—	—	—	—	—	—	—
140	—	—	—	—	—	—	—

TABLE 33

AMERICAN "ANNULAR BEARING ENGINEERS' COMMITTEE" BALL AND ROLLER BEARING LIMITS, CLASS IV

(Tolerance unit = 0·001 in.)

Bearing Bore, mm	Outside Dia. − 0			Bearing Bore − 0	Eccentricity				Width − 0
	Light	Med.	Heavy		Inner Ring, max.	Outer Ring, max. Light	Med.	Heavy	
10	− 0·2	− 0·3	—	− 0·2	0·2	0·3	0·4	—	− 2·0
12	− 0·3	− 0·3	—	− 0·2	0·2	0·4	0·4	—	− 2·0
15	− 0·3	− 0·3	—	− 0·2	0·2	0·4	0·4	—	− 2·0
17	− 0·3	− 0·3	− 0·3	− 0·2	0·2	0·4	0·4	0·4	− 2·0
20	− 0·3	− 0·3	− 0·3	− 0·2	0·2	0·4	0·4	0·4	− 2·0
25	− 0·3	− 0·3	− 0·3	− 0·2	0·2	0·4	0·4	0·4	− 2·0
30	− 0·3	− 0·3	− 0·4	− 0·2	0·2	0·4	0·4	0·5	− 2·0
35	− 0·3	− 0·3	− 0·4	− 0·3	0·3	0·4	0·4	0·5	− 2·0
40	− 0·3	− 0·4	− 0·4	− 0·3	0·3	0·4	0·5	0·5	− 2·0
45	− 0·4	− 0·4	− 0·4	− 0·3	0·3	0·5	0·5	0·5	− 2·0
50	− 0·4	− 0·4	− 0·4	− 0·3	0·3	0·5	0·5	0·6	− 2·0
55	− 0·4	− 0·4	− 0·4	− 0·3	0·3	0·5	0·5	0·6	− 2·0
60	− 0·4	− 0·4	− 0·5	− 0·3	0·3	0·5	0·6	0·6	− 2·0
65	− 0·4	− 0·4	− 0·5	− 0·3	0·3	0·5	0·6	0·6	− 2·0
70	− 0·4	− 0·5	− 0·5	− 0·3	0·3	0·6	0·6	0·6	− 2·0
75	− 0·4	− 0·5	− 0·6	− 0·3	0·3	0·6	0·6	0·8	− 2·0
80	− 0·4	− 0·5	− 0·6	− 0·3	0·3	0·6	0·6	0·8	− 2·0
85	− 0·5	− 0·5	− 0·6	− 0·4	0·4	0·6	0·6	0·8	− 3·0
90	− 0·5	− 0·6	− 0·6	− 0·4	0·4	0·6	0·8	0·8	− 3·0
95	− 0·5	− 0·6	− 0·6	− 0·4	0·4	0·6	0·8	0·8	− 3·0
100	− 0·5	− 0·6	− 0·6	− 0·4	0·4	0·6	0·8	0·8	− 3·0
105	− 0·6	− 0·6	—	− 0·4	0·4	0·8	0·8	—	− 3·0
110	− 0·6	− 0·6	—	− 0·4	0·4	0·8	0·8	—	− 3·0
120	− 0·6	− 0·6	—	− 0·4	0·4	0·8	0·8	—	− 3·0
130	− 0·6	− 0·7	—	− 0·5	0·5	0·8	1·0	—	− 3·0
140	− 0·6	− 0·7	—	− 0·5	0·5	0·8	1·0	—	− 3·0

TABLE 34

Limits on Timken Tapered Roller Bearings

A. INCH Bearings and Non-standard Metric Bearings (Unit 0·001 in.)

Dia., in.	Normal		Precision	
	Bore	O.D.	Bore	O.D.
– 3·187	+ 0·6 + 0	+ 0·6 + 0	+ 0·5 + 0	+ 0·5 + 0
3·19–11·99	+ 1·0 + 0	+ 1·0 + 0	+ 0·5 + 0	+ 0·5 + 0
12·0 –14·5	+ 2·0 + 0	+ 2·0 + 0	+ 0·5 + 0	+ 0·5 + 0
14·6 –24	+ 2·0 + 0	+ 2·0 + 0	—	—

B. METRIC I.S.A. Bearings (Unit 0·001 mm)

Dia., mm	Normal		Precision	
	Bore	O.D.	Bore	O.D.
10·1– 30	– 0 – 10	– 0 – 8	– 0 – 10	– 0 – 8
30·1– 50	– 0 – 13	– 0 – 10	– 0 – 13	– 0 – 10
50·1– 80	– 0 – 15	– 0 – 13	– 0 – 13	– 0 – 13
80·1–120	– 0 – 20	– 0 – 15	– 0 – 13	– 0 – 13
120·1–150	—	– 0 – 18	—	– 0 – 13

C. ALL Bearings—Eccentricity and Axial Wobble

Dia., in.	Normal		Precision		Extra Precision	
	Eccent.	Wobble	Eccent.	Wobble	Eccent.	Wobble
– 8	0·0008	0·001	0·0005	0·001	0·0003	0·0005
8·01–14·5	0·0012	0·0015	0·0008	0·0015	0·0003	0·0008

The Standard Class I tolerances are for normal use. Class II bearings have closer bore tolerances, while Class III bearings have

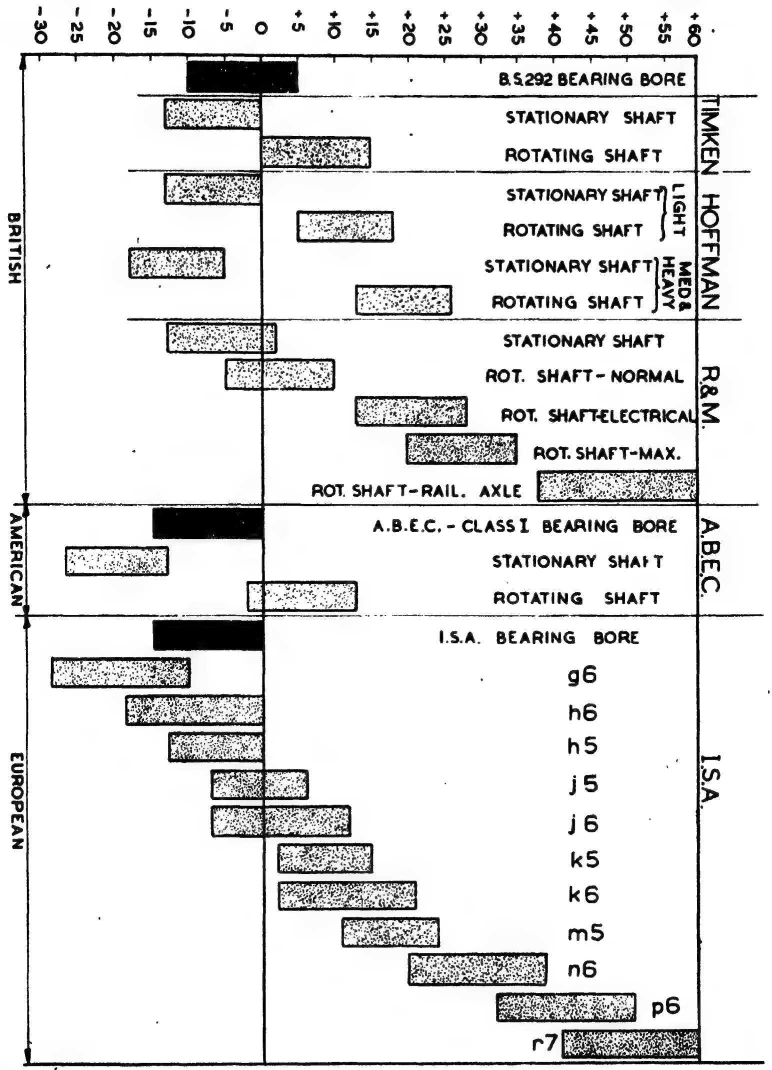


FIG. 78. BALL BEARING FITS SHOWN DIAGRAMMATICALLY—SHAFTS

closer outside diameter tolerances, the same bore limits as Class II, and closer eccentricity limits on the inner race. Class IV tolerances are for super-precision work.

2. Installation Limits for Ball and Roller Bearings

The selection of limits for ball and roller bearing shaft and housing fits is a complicated procedure requiring much experience for the best results. It is not a subject that can easily be dealt with

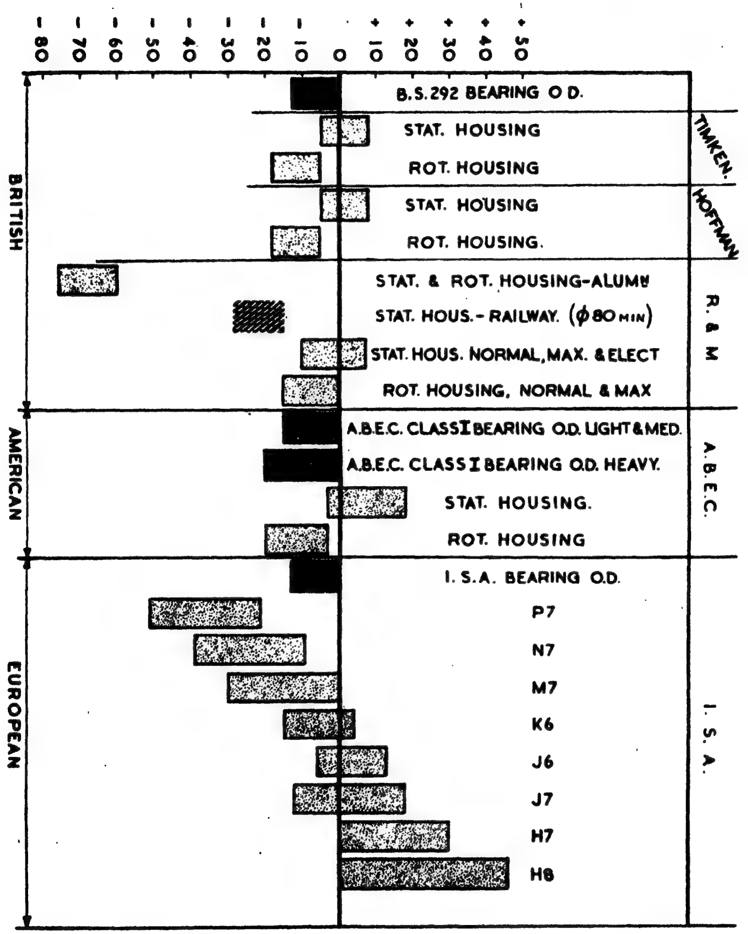


Fig. 79. Ball Bearing Fits shown Diagrammatically—Housings

comprehensively in a short chapter, and the reader is advised to consult the bearing makers in case of doubt arising. The bearing makers, however, do not make matters more simple, as already mentioned, by agreeing amongst themselves. A glance at the tables and diagrams will make this very clear.

Perhaps the most comprehensive information is that published by the S.K.F. Company in several countries. As they were very largely involved in the drafting of the I.S.A. bearing standards and relevant sections of the I.S.A. tolerance system, they have undoubtedly had occasion to consider the problem fully. Their recommendations are therefore given in full.

Figs. 78 and 79 show the various makers' fits compared diagrammatically.

The main factors which have to be considered, or affect the performance of a bearing, are as follows—

(1) The amount of slackness in the bearing itself, usually desirable to some degree.

(2) The change in this slackness due to fit interference (not applicable to adjustable tapered roller bearings).

(3) The change in this slackness due to differential temperature.

(4) The rigidity of bearing races, and the effect of housing or shaft truth errors.

(5) The use of sufficient fit interference to prevent "epicyclic" creep.

(6) Or, conversely, the use of just sufficient clearance to permit axial adjustment of shaft or housing.

(7) The effect of the applied load on the fits.

(8) The stiffness or elasticity of shaft or housing materials, particularly aluminium alloys as compared with steel.

Tables

In the diagrammatic representation of the fits recommended by the various makers the very wide differences are clearly shown. It must be admitted that it is sometimes difficult to know what the poor designer should do in the face of so much conflicting evidence. For the best work the I.S.A.—S.K.F. proposals appear to be the most satisfactory, although for many designs other makers' recommendations will be used.

Where export designs are considered, the availability of replacement bearings overseas made to I.S.A. or A.B.E.C. limits should be considered with the consequent effect on bearing fits.

Table 35	gives	S.K.F.	shaft fits
Table 36	,,	,,	housing fits
Table 37	,,	,,	special limits for inch bearings
Table 38	,,	,,	thrust bearing fits
Table 39	,,	other	shaft fits: inch system
Table 40	,,	,,	housing fits: inch system
Table 41	,,	,,	shaft fits: metric system

TABLE [illegible]

S.K.F. Shaft Limits for Radial Bearings

	Conditions	Examples	Shaft. Dia. in mm and in. Ball Bearings	Cylindrical and Taper Roller Bearings	Spherical Roller Bearings	Tolerance symbol *	Remarks
Bearings with Cylindrical Bore							
Stationary inner ring load	Easy axial displacement of inner ring on shaft desirable	Wheels on non-rotating axles	Metric shafts Inch shafts			g6 g6i	
	Easy axial displacement of inner ring on shaft unnecessary	Tension pulleys; rope sheaves	Metric shafts Inch shafts			h6 h6i	
Rotating inner ring load, or direction of loading indeterminate	Light and variable loads	Electrical apparatus; machine tools; pumps; ventilating fans; transport vehicles	≤ 18 ¾"			h5 h5i	By light loads is understood such as are, as a rule, no greater than 6–7% of the specific dynamic capacity. For very accurate bearing arrangements, j5, k5, and m5 are used instead of j6, k6, and m6
			18–100 ¾"–4"	≤ 40 1½"	≤ 40	j6 j6i	
			> 100 4"	40–160 1½"–6"	40–100	k6 k6i	
				160–225 6"–9"	100–225	m6 m6i	
	Normal and heavy loads	Bearing arrangements generally; electric motors; turbines; pumps; internal combustion engines; gearing; woodworking machines	≤ 18 ¾"			j5 j5i	For taper roller bearings, k6 or m6 can be used as a rule instead of k5 or m5, for in the fitting of bearings of this type there is no need to take the decrease in bearing slackness into account
			18–100 ¾"–4"	≤ 40 1½"	≤ 40	k5 k5i	
			> 100 4"	40–160 1½"–6"	40–100	m5 m5i	
				160–225 6"–9"	100–200	n6 n6i	
				225–400	200–355	p6	
					355–500	r6	
					> 500	r7	
	Very heavy loads; shock loads	Axleboxes for locomotives and other heavy railway vehicles; traction motors		60–160 2⅜"–6"	60–100	n6 n6i	More than the normal bearing slackness necessary
				160–225 6"–9"	100–200	p6 p6i	
					200–355	r6	
					355–500	r7	
	Purely axial load	Bearing arrangements of all kinds	Metric shafts Inch shafts			j6 j6i	
Bearings with Taper Bores and Taper Sleeves							
Loads of all kinds		Bearing arrangements in general; railway axleboxes	Metric and inch shafts			h9/IT5	The symbols IT5 and IT7 following the tolerance symbol mean that the deviations of the shaft from its true geometric form, i.e. out-of-round and taper, must be no greater than the tolerances of grades 5 and 7 respectively
		Lineshafting	Metric and inch shafts			h10/IT7	

* The tolerances g6, h6, h5, etc., are used for metric shafts.
The tolerances g6i, h6i, h5i, etc., are used for inch shafts.
The values represented by the tolerance symbols are given in Table 37.

TABLE 36

S.K.F. Cast Iron or Steel* Housing Limits for Radial Bearings

Conditions			Examples	Tolerance symbol†	Remarks‡
Solid housings	Rotating outer ring load	Heavy loads on bearings in thin-walled housings; heavy shock-loads	Roller bearing wheel-hubs; big-end bearings; crane runner wheels	P7 P7i	Outer ring cannot be displaced
		Normal and heavy loads	Ball bearing wheel-hubs; big-end bearings	N7 N7i	
		Light and variable loads	Conveyor rollers; rope sheaves; belt-tension pulleys	M7 M7i	
	Direction of loading indeterminate	Heavy shock-loads	Electric traction motors		
		Heavy and normal loads; axial mobility of outer ring unnecessary	Electric motors; pumps; crank-shaft main bearings	K7 K7i	Outer ring cannot as a rule be displaced
Split or solid housings		Normal and light loads; axial mobility of outer ring desirable	Electric motors; pumps; crank-shaft main bearings	J7 J7i	Outer ring can as a rule be displaced
	Stationary outer ring load	Shock loads; occasional complete unloading	Railway axleboxes		
		All loads	Bearing arrangements generally	H7 H7i	Outer ring easily displaced
		Normal and light loads, together with simple working conditions	Lineshafting	H8 H8i	
		Heat conduction through walls of hollow shafts	Drying cylinders	G7 G7i	
Solid housings	Bearing arrangement extra accurate	Very true running and great rigidity under variable load	Roller bearings for machine-tool main spindles: $D > 250$ mm $D = 125$–250 mm $D \leq 125$ mm $D > 10$ in. $D = 5$–10 in. $D \leq 5$ in.	P6 N6 M6 P6i N6i M6i	Outer ring cannot be displaced
		Very true running under light loads of indeterminate direction	Ball bearings at the work end of grinding spindles; locating bearings in high-speed centrifugal compressors	K6 K6i	Outer ring cannot as a rule be displaced
		Very true running; axial mobility of outer ring desirable	Ball bearings at drive end of grinding spindles; axially-free bearings in high-speed centrifugal compressors	J6 J6i	Outer ring can be displaced

* For light alloy housings, limits are generally used that give a somewhat tighter fit than is obtained with the limits specified above.

† The tolerances P7, N7, M7, etc., are used for metric housings.
The tolerances P7i, N7i, M7i, etc., are used for inch housings.
The limits represented by the tolerance symbols are given in Table 37.

‡ These remarks indicate whether the fit allows non-separable bearings to have axial freedom or not.

TABLE 37. S.K.F. Special Limits

A. Shafts

Nominal Shaft Dia., in.	g6i	h6i	h5i	j5i	j6i	k5i	k6i
0·126– 0·375	– 0·1, – 0·4	+ 0·1, – 0·2	+ 0·1, – 0·1	+ 0·3, + 0·1	—	—	—
0·376– 0·75	– 0·1, – 0·5	+ 0·1, – 0·3	+ 0·1, – 0·2	+ 0·3, + 0	+ 0·4, + 0	+ 0·5, + 0·2	+ 0·6, + 0·2
0·751– 1·125	– 0·1, – 0·6	+ 0·1, – 0·4	+ 0·1, – 0·2	+ 0·3, + 0	+ 0·5, + 0	+ 0·5, + 0·2	+ 0·7, + 0·2
1·126– 2	– 0·1, – 0·7	+ 0·2, – 0·4	—	+ 0·4, + 0	+ 0·6, + 0	+ 0·7, + 0·3	+ 0·9, + 0·3
2·01 – 3·125	– 0·1, – 0·9	+ 0·3, – 0·5	—	+ 0·5, – 0·1	+ 0·7, – 0·1	+ 0·9, + 0·3	+ 1·1, + 0·3
3·13 – 4·75	– 0, – 1·1	+ 0·5, – 0·6	—	+ 0·7, – 0·1	+ 1·0, – 0·1	+ 1·1, + 0·3	+ 1·4, + 0·3
4·76 – 7	– 0, – 1·2	+ 0·5, – 0·7	—	—	+ 1·0, – 0·2	+ 1·3, + 0·3	+ 1·5, + 0·3
7·01 –10	– 0, – 1·6	+ 0·7, – 0·9	—	—	+ 1·3, – 0·3	+ 1·6, + 0·4	+ 2·0, + 0·4
Fit of Inner Ring on Shaft	Sucking Fit	Sucking to Light Driving Fit	Light Driving Fit	Driving Fit		Light Force Fit	

* The ovality, taper, etc., of shaft seatings, with tolerances h9 or h10 for adaptor or withdrawal sleeve

B. Housings

Nominal Bore, in.	G7i	H8i	H7i	J7i	J6i	K6i
– 0·75	+ 0·6, – 0·1	+ 0·8, – 0·3	+ 0·4, – 0·3	+ 0·1, – 0·6	– 0, – 0·4	– 0·2, – 0·6
0·751–1·125	+ 0·7, – 0·1	+ 0·9, – 0·3	+ 0·5, – 0·3	+ 0·1, – 0·7	– 0, – 0·5	– 0·2, – 0·7
1·126– 1·875	+ 0·9, – 0·1	+ 1·2, – 0·3	+ 0·7, – 0·3	+ 0·2, – 0·8	– 0, – 0·6	– 0·2, – 0·8
1·88 – 2·875	+ 1·1, – 0·1	+ 1·2, – 0·5	+ 0·7, – 0·5	+ 0·2, – 1·0	– 0, – 0·7	– 0·3, – 1·0
2·88 – 4·75	+ 1·1, – 0·2	+ 1·2, – 0·7	+ 0·6, – 0·7	+ 0·2, – 1·1	– 0·1, – 0·9	– 0·5, – 1·3
4·76 – 7	+ 1·2, – 0·3	+ 1·3, – 1·0	+ 0·5, – 1·0	+ 0·2, – 1·3	– 0·3, – 1·3	– 0·8, – 1·8
7·01 –10	+ 1·8, – 0·3	+ 2·2, – 1·0	+ 1·1, – 1·0	+ 0·5, – 1·6	—	—
10·01 –12	+ 2·0, – 0·3	+ 2·5, – 1·0	+ 1·3, – 1·0	+ 0·7, – 1·6	—	—
12·1 –16	+ 2·2, – 0·3	+ 2·8, – 1·0	+ 1·5, – 1·0	+ 0·8, – 1·7	—	—
Fit of Outer Ring in Housing	Accurate Sliding Fit	Push Fit		Sucking Fit		Light Driving Fit

or B.S. 292 Inch Bearings

nit = 0·001 in.)

m5i	m6i	n6i	p6i	h9*	IT5	h10*	IT7
—	—	—	—	—	—	—	—
—	—	—	—	- 0, - 1·7	0·3	- 0, - 2·8	0·7
—	—	—	—	- 0, - 2·0	0·4	- 0, - 3·3	0·8
+ 1·0, + 0·6	—	—	—	- 0, - 2·4	0·5	- 0, - 3·9	1·0
+ 1·2, + 0·6	+ 1·4, + 0·6	+ 1·8, + 1·0	+ 2·3, + 1·5	- 0, - 2·9	0·5	- 0, - 4·7	1·2
+ 1·5, + 0·7	+ 1·8, + 0·7	+ 2·2, + 1·1	+ 2·8, + 1·6	- 0, - 3·4	0·6	- 0, - 5·5	1·4
+ 1·8, + 0·8	+ 2·0, + 0·8	+ 2·5, + 1·3	+ 3·2, + 1·9	- 0, - 3·9	0·7	- 0, - 6·3	1·6
+ 2·1, + 0·9	+ 2·5, + 0·9	+ 3·0, + 1·4	+ 3·7, + 2·2	- 0, - 4·5	0·8	- 0, - 7·3	1·8
Force Fit		Hard Force Fit		The limits are for bearings with adaptor or withdrawal sleeves			

earings must not exceed the tolerance IT5 or IT7 respectively; note that these limits are S.K.F.'s translations.

nit = 0·001 in.)

K7i	M6i	M7i	N6i	N7i	P6i	P7i
- 0·1, - 0·8	- 0·5, - 0·9	- 0·3, - 1·0	—	—	—	—
- 0·1, - 0·9	- 0·5, - 1·0	- 0·3, - 1·1	—	- 0·6, - 1·4	—	—
- 0·1, - 1·1	- 0·5, - 1·1	- 0·3, - 1·3	—	- 0·6, - 1·6	—	- 1·0, - 2·0
- 0·1, - 1·3	- 0·7, - 1·4	- 0·5, - 1·7	—	- 0·8, - 2·0	—	- 1·3, - 2·5
- 0·3, - 1·6	- 0·9, - 1·7	- 0·7, - 2·0	—	- 1·1, - 2·4	—	- 1·6, - 2·9
- 0·5, - 2·0	- 1·3, - 2·3	- 1·0, - 2·5	- 1·8, - 2·8	- 1·4, - 2·9	—	- 2·0, - 3·5
- 0·2, - 2·3	—	- 0·7, - 2·8	- 1·5, - 3·0	- 1·3, - 3·4	- 2·2, - 3·7	- 2·0, - 4·1
- 0·1, - 2·4	—	- 0·7, - 3·0	- 1·6, - 3·2	- 1·3, - 3·6	- 2·5, - 4·1	- 2·1, - 4·4
- 0, - 2·5	—	- 0·7, - 3·2	- 1·7, - 3·4	- 1·3, - 3·8	- 2·7, - 4·4	- 2·3, - 4·8
Light Driving Fit	Driving Fit		Hard Driving Fit		Light Force Fit	

Table 42 gives other housing fits: metric system
Table 43 ,, Timken tapered roller bearing: shaft fits, inch system
Table 44 ,, Timken tapered roller bearing: housing fits, inch system
Table 45 ,, Timken tapered roller bearing: shaft fits, metric system
Table 46 ,, Timken tapered roller bearing: housing fits, metric system.

3. Needle Bearings

The following information is based on Ransome and Marles needle bearing practice; American Bantam recommendations are similar unless indicated otherwise. The installation problems of needle bearing assemblies can be regarded as similar to normal ball-bearing practice, except that heavy interference fits should be avoided. If, however, the needles are to be used on specially made race tracks, as is often the case, the special limits given below are applicable.

The Bantam Company recommends the maximum interference fits given in Table 47.

Needle bearings must run with appreciable diametral clearance as shown in Table 48.

It has been found that in practice almost the whole of any interference on shaft or housing will be transferred to the race dimensions, decreasing the diametral clearance. Good practice would be to add to the nominal clearance the maximum interference of both fits, less, say, 25 per cent, to allow for the improbability of both fits being on maximum interference together.

The limits on the needles themselves (diameter 2·5–5 mm) are $\pm$ 0·0001 in. ($\pm$ 0·0025 mm). The recommended limits to be applied to the tracks are—

To nominal diameter of inner track: $-$ 0, $-$ 0·0005 in.
($-$ 0, $-$ 0·013 mm)

To nominal diameter of outer track: $+$ 0·0007, $+$ 0·0002 in.
($+$ 0·018, $+$ 0·005 mm)

Width of shouldered track: Nominal roller length
$+$ 0·012, $+$ 0·008 in.
($+$ 0·3, $+$ 0·2 mm)

TABLE 38*

S.K.F. THRUST BEARING FITS—METRIC BEARINGS

SHAFTS

Conditions		Shaft dia. in mm	Fit
Pure Thrust Load		All diameters	j6
Combined load on spherical roller thrust bearings.	Fixed load on inner ring.	All diameters	j6
	Rotating load on inner ring or indeterminate load direction.	≤ 200	k6
		200 . . . 400	m6
		> 400	n6

HOUSINGS

Conditions		Fit	Remarks
Pure thrust load	Thrust ball bearings.	H8	In less accurate applications, the housing ring or spherical washer respectively are mounted with radial play.
	Spherical roller thrust bearing; companion bearing to carry radial load.	--	The outer ring is mounted with radial play.
Combined load on spherical roller thrust bearings.	Fixed load on outer ring.	J7	
	Rotating load on outer ring or indeterminate load direction.	K7	In general.
		M7	In case of comparatively heavy radial load.

* Reproduced from *Ball and Roller Bearing Engineering*, by A. Palmgren.

TABLE 39

Various Makers' Shaft Fits: Inch System

A. Stationary Shaft (Unit = 0·001 in.)

Shaft Dia., in.	Timken (not tapered)	Hoffman		R. & M.
		Light	Medium, Heavy	Normal, Maximum
0 – 2	− 0 − 0·4	− 0 − 0·5	− 0 − 0·5	+ 0·1 − 0·4
2·01– 3	− 0 − 0·4*	− 0 − 0·5	− 0·2 − 0·7	+ 0·1 − 0·4
3·01– 4	− 0 − 0·5*†	− 0 − 0·5	− 0·2 − 0·7	− 0 − 0·5
4·01– 5	− 0 − 0·6*†	− 0·2 − 0·7	− 0·5 − 1·0	− 0 − 0·6
5·01– 6	− 0 − 0·7*†	− 0·2 − 0·7	− 0·5 − 1·0	− 0 − 0·7
6·01–12	− 0 − 1·0†	− 0·2 − 0·7	− 0·5 − 1·0	− 0 − 0·9

* excludes 3, 4, 5, and 6 in. † includes 3, 4, 5, and 6 in.

B. Rotating Shaft (Unit = 0·001 in.)

Shaft Dia., in.	Timken (not tapered)	Hoffman		Ransome & Marles				
		Light	Med., Heavy	Normal	Electrical	Dia., in.	Max. Limits	Railway Axle Limits
0 – 2·0	+ 0·6	+ 0·7	+ 0·7	+ 0·4	+ 0·6	0 –0·749	+ 0·9 + 0·4	—
	+ 0	+ 0·2	+ 0·2	− 0·1	+ 0·2	0·75–1·24	+ 1·0 + 0·5	—
2·01– 2·99	+ 0·6	+ 0·7	+ 1·0	+ 0·4	+ 1·0	1·25–1·74	+ 1·1 + 0·6	—
	+ 0	+ 0·2	+ 0·5	− 0·1	+ 0·5	1·75–2·49	+ 1·3 + 0·8	—
3 – 3·99	+ 0·6 + 0	+ 0·7* + 0·2	+ 1·0* + 0·5	+ 0·4* − 0·1	+ 1·0* + 0·5	2·5 – 3·49	+ 1·5 + 1·0	+ 2·5 + 1·5
4 – 4·99	+ 0·6 + 0	+ 1·0† + 0·5	+ 1·2† + 0·7	+ 0·4† − 0·1	+ 1·2† + 0·7	3·5 – 4·74	+ 1·7 + 1·2	+ 2·7 + 1·7
5 – 5·99	+ 0·8 + 0	+ 1·0 + 0·5	+ 1·2 + 0·7	+ 0·4‡ − 0·1	+ 1·2 + 0·7	4·75– 6	+ 1·9 + 1·4	+ 2·9 + 1·9
6 –11·99	+ 1·0 + 0	+ 1·0 + 0·5	+ 1·2 + 0·7	+ 0·4§ − 0·3	—	6·01–12	+ 2·3 + 1·6	—

* includes 4·0; † excludes 4·0; ‡ includes 6·0; § excludes 6·0 in.

TABLE 40

VARIOUS MAKERS' HOUSING FITS: INCH SYSTEM

A. STATIONARY HOUSING (Unit = 0·001 in.)

Housing Dia., in.	Timken (not tapered)	Hoffman	Ransome & Marles			
			Normal, Maximum	"Aluminium"	"Electrical"	Railway Axle
0– 1·99	– 0 – 0·5	– 0 – 0·5	– 0 – 0·7	– 2·8 – 3·3	– 0 – 0·5	—
2– 2·99	– 0·2 – 0·7	– 0·2 – 0·7	– 0·2 – 0·9	– 3·0 – 3·5	– 0·1 – 0·7	—
3– 4·99	– 0·5 – 1·0	– 0·5 – 1·0	– 0·4 – 1·2	– 3·3 – 4·0	– 0·1 – 0·9	– 1·1 – 1·9
5–11·99	– 0·8 – 1·5	– 0·8 – 1·3	– 0·8 – 1·8	– 3·8 – 4·8	– 0·3 – 1·3	– 1·4 – 2·4
12–19·9	– 0·8 – 1·8	– 0·8 – 1·3	– 0·8 – 2·1	– 4·3 – 5·3	—	– 1·5 – 2·8

B. ROTATING HOUSING (Unit = 0·001 in.)

Housing Dia., in.	Timken (not tapered)	Hoffman	Ransome & Marles	
			Normal, Maximum	"Aluminium"
0– 1·99	– 0·5 – 1·0	– 0·5 – 1·0	– 0·3 – 1·0	– 2·8 – 3·3
2– 2·99	– 0·7 – 1·2	– 0·7 – 1·2	– 0·5 – 1·2	– 3·0 – 3·5
3– 4·99	– 1·0 – 1·5	– 1·0 – 1·5	– 0·7 – 1·5	– 3·3 – 4·0
5–11·99	– 1·3 – 2·0	– 1·5 – 2·0	– 1·2 – 2·1	– 3·8 – 4·8
12–19·99	– 1·3 – 2·4	– 1·5 – 2·0	– 1·2 – 2·4	– 4·3 – 5·3

TABLE 41

Various Makers' Shaft Fits: Metric System

A. Stationary Shaft (Unit = 0·001 mm)

Shaft Dia., mm	Timken (not tapered)	Hoffmann Light	Hoffmann Medium, Heavy	R. & M. Normal, Maximum	A.B.E.C. Class I Dia., mm	A.B.E.C. Class I Limits (converted)
0 – 50	− 0	− 0	− 0	+ 2	10–17	− 5 − 13
	− 13	− 13	− 13	− 13	20–30	− 7 − 18
50·1– 79·9	− 0	− 0	− 5	+ 2*	30–50	− 10 − 23
	− 13	− 13	− 18	− 13	55–80	− 13 − 27
80 –100	− 5	− 0	− 5	− 0†	85–120	− 18 − 35
	− 18	− 13	− 18	− 18		
100·1–139·9	− 5	− 5	− 13	− 0*	130, 140	− 23 − 45
	− 18	− 18	− 26	− 18		
140 –199·9	− 5	− 5	− 13	− 2†		
	− 20	− 18	− 26	− 20		

* includes 80 and 140 mm; † excludes 80 and 140 mm.

B. Rotating Shaft (Unit = 0·001 mm)

Shaft Dia., mm	Timken (not tapered)	Hoffman Light	Hoffman Med. Heavy	Ransome & Marles Normal	Ransome & Marles Electrical	Ransome & Marles Dia., mm	Ransome & Marles Maximum Limits	Ransome & Marles Railway Axle Limits	A.B.E.C. Class I Dia., mm	A.B.E.C. Class I Limits (converted)
0 – 50	+ 15	+ 18	+ 18	+ 10	+ 10	0 – 19·9	+ 21 + 6	—	10–17	+ 5 − 3
	+ 0	+ 5	+ 5	− 5	+ 5	20 – 29·9	+ 24 + 9	—	20–30	+ 7 − 3
50·1– 79·9	+ 15	+ 18	+ 26	+ 10*	+ 28*	30 – 44·9	+ 27 + 12	—	30–50	+ 10 − 3
	+ 0	+ 5	+ 13	− 5	+ 13	45 – 59·9	+ 31 + 16	—	55–80	+ 13 − 3
80 –100	+ 15	+ 18	+ 26	+ 10†	+ 27†	60 – 80	+ 35 + 20	+ 60 + 38	85–120	+ 15 − 3
	+ 0	+ 5	+ 13	− 8	+ 13					
100·1–139·9	+ 15	+ 26	+ 31	+ 10*	+ 33*	80·1–120	+ 41 + 23	+ 64 + 42	130, 140	+ 20 − 3
	+ 0	+ 13	+ 18	− 8	+ 18					
140 –199·9	+ 25	+ 26	+ 31	+ 12†	+ 33†	120·1–200	+ 44 + 26	+ 69 + 47		
	+ 0	+ 13	+ 18	− 8	+ 18					

* includes 80 and 140 mm; † excludes 80 and 140 mm.

TABLE 42

Various Makers' Housing Fits: Metric System

A. Stationary Housing (Unit = 0·001 mm)

Housing Dia., mm	Timken (not tap-ered)	Hoff-man	Ransome & Marles				A.B.E.C. Class I	
			Normal, Max.	"Alum-inium"	"Elec-trical"	Rail-way Axle	Dia., mm	Limits (con-verted)
0– 74·9	+ 8 − 5	+ 8 − 5	+ 7* − 10	− 60* − 76	+ 7‡ − 5	—	−30 32–47	+ 10 − 3 + 12 − 3
75–129·9	+ 10 − 5	+ 10 − 3	+ 10† − 15	− 60† − 82	+ 10§ − 5	− 15† − 28	52–80	+ 18 − 3
130–149·9	+ 10 − 5	+ 13 + 0	+ 10* − 15	− 60* − 82	+ 10 − 5	− 15* − 28	85–120	+ 20 − 5
150–200	+ 13 − 10	+ 13 + 0	+ 10† − 20	− 65† − 92	+ 13 − 5	− 15† − 28	125–180	+ 25 − 5

* includes 75 and 150 mm; † excludes 75 and 150 mm; ‡ up to 50; § from 50·1.

B. Rotating Housing (Unit = 0·001 mm)

Housing Dia., mm	Timken (not tapered)	Hoffman	Ransome & Marles		A.B.E.C. Class I	
			Normal, Maximum	"Alu-minium"	Dia., mm	Limits (converted)
0– 74·9	− 5 − 18	− 5 − 18	− 0* − 15	− 60* − 76	− 30 32– 47	− 3 − 15 − 3 − 18
75–129·9	− 5 − 18	− 5 − 18	− 0† − 22	− 60† − 82	52– 80	− 3 − 20
130–149·9	− 5 − 18	− 5 − 18	− 0* − 22	− 60* − 82	85–120	− 3 − 25
150–200	− 5 − 25	− 5 − 18	− 0† − 27	− 65† − 92	125–180	− 3 − 32

* includes 75 and 150 mm; † excludes 75 and 150 mm.

TABLE 43

Timken Tapered Roller Bearing Shaft Fits: Inch System

A. Stationary Shaft (Unit = 0·001 in.)

Industry	Type of Application	Cones	Bearing Bore, in.		
			–0·624	0·625–3·1875	3·19–6
Automotive	Front wheels, fully floating rear wheels, etc.	Adjust.	—	− 0·2 − 0·7	− 0·2 − 1·2
Industrial	Machine tool spindles, precision bearings.	Adjust.	+ 0·5 + 0	+ 0·5 + 0	+ 0·5 + 0
		Non-adj.	+ 0·5 + 0	+ 1·5 + 1·0	+ 1·5 + 1·0
	Transmissions, speed reducers, electrical equipment, etc.	Adjust.	+ 0·5 + 0	+ 0·5 + 0	+ 1·0 + 0
		Non-adj.	+ 0·5 + 0	+ 1·5 + 1·0	+ 2·5 + 1·5
	Low speed, heavy duty, continuous service, shock, etc.	Adjust.	—	+ 2·5 + 1·5	—
		Non-adj.	—	+ 2·5 + 1·5	—
	Sheaves, crown blocks, front wheels, implement wheels, etc.	Adjust.	—	− 0 − 0·5	− 0 − 1·0

B. Rotating Shaft (Unit = 0·001 in.)

Industry	Type of Application	Cones	Bearing Bore, in.		
			–0·624	0·625–3·1875	3·19–6
Automotive	Worm shaft, pinion, differential, transmission, rear wheels, cross shaft.	Adjust.	—	+ 0·5 + 0	+ 1·5 + 0·5
		Non-adj.	—	+ 1·5 + 1·0	+ 2·5 + 1·5
Industrial	Machine tool spindles, precision bearings.	Adjust.	+ 0·5 + 0	+ 1·5 + 1·0	+ 1·5 + 1·0
		Non-adj.	+ 0·5 + 0	+ 1·5 + 1·0	+ 1·5 + 1·0
	Transmissions, speed reducers, electrical equipment, etc.	Adjust.	+ 0·5 + 0	+ 1·5 + 1·0	+ 2·5 + 1·5
		Non-adj.	+ 0·5 + 0	+ 1·5 + 1·0	+ 2·5 + 1·5
	Low speed, heavy duty, continuous service, shock, etc.	Adjust.	—	+ 2·5 + 1·5	—
		Non-adj.	—	+ 2·5 + 1·5	—

TABLE 44

Timken Tapered Roller Bearing Housing Fits: Inch System

(Unit = 0·001 in.)

Industry	Type of Application	Cup	Bearing Outside Dia., in.		
			–2·99	3·0–6·0	6·01–12
Automotive	Front wheels, fully floating rear wheels, pinion, differential.	Non-adj.	− 0·5 − 1·5	− 1·0 − 2·0	− 1·0 − 3·0
	Rear wheels (private cars), transmission, cross shaft, other applications.	Adjust.	+ 0·6 + 0	− 0 − 1·0	+ 2·0 + 0
Industrial	Machine tool spindles, precision bearings.	Adjust.	*	*	*
		Non-adj.	− 0·5 − 1·0	− 0·5 − 1·0	− 0·5 − 1·5
		Floating	*	*	*
	Transmission, speed reducers, electrical machinery, sheaves, crown blocks, low speed, heavy duty, continuous service, shock loads.	Adjust.	+ 1·0 + 0	+ 1·0 + 0	+ 2·0 + 0·5
		Non-adj.	− 0·5 − 1·5	− 1·0 − 2·0	− 1·0 − 3·0
		Floating	+ 2·0 + 1·0	+ 2·0 + 1·0	+ 3·0 + 2·0
	Tractor front wheels, implement wheels, etc.	Non-adj.	− 0·5 − 1·5	− 1·0 − 3·0	− 1·0 − 3·0

* By selection.

TABLE 45

Timken Tapered Roller Bearing Shaft Fits: Metric I.S.A. Bearings

A. Stationary Shaft (Unit = 0·001 mm)

Industry	Type of Application	Cones	Shaft Dia., mm			
			-30	30·1–50	50·1–80	80·1–120
Automotive	Front wheels, fully floating rear wheels, tractor wheels, etc.	Adjust.	− 20 − 33	− 25 − 50	− 28 − 53	− 33 − 58
Industrial	Machine tool spindles, precision bearings.	Adjust.	+ 3 − 10	+ 3 − 10	+ 3 − 10	+ 3 − 10
		Non-adj.	+ 3 − 10	+ 25 + 13	+ 25 + 13	+ 25 + 13
	Transmissions, speed reducers, electrical equipment, etc.	Adjust.	+ 3 − 10	− 0 − 13	− 0 − 13	+ 13 − 13
		Non-adj.	+ 23 + 10	+ 25 + 13	+ 25 + 13	+ 38 + 13
	Low speed, heavy duty, continuous service, shock, etc.	Adjust.	—	+ 45 + 20	+ 50 + 25	+ 55 + 30
		Non-adj.	—	+ 45 + 20	+ 50 + 25	+ 55 + 30
	Sheaves, crown blocks, front wheels, implement wheels, etc.	Adjust.	− 10 − 23	− 13 − 25	− 15 − 28	− 20 − 45

B. Rotating Shaft (Unit = 0·001 mm)

Industry	Type of Application	Cones	Shaft Dia., mm			
			-30	30·1–50	50·1–80	80·1–120
Automotive	Worm shaft, pinion, differential, transmission, rear wheels, cross shaft.	Adjust.	+ 3 − 10	− 0 − 13	− 0 − 13	+ 13 − 13
		Non-adj.	+ 23 + 10	+ 25 + 13	+ 25 + 13	+ 38 + 13
Industrial	Machine tool spindles, precision bearings.	Adjust.	+ 8 − 5	+ 25 + 13	+ 25 + 13	+ 25 + 13
		Non-adj.	+ 8 − 5	+ 25 + 13	+ 25 + 13	+ 25 + 13
	Transmission, speed reducers, electrical equipment, etc.	Adjust.	+ 23 + 10	+ 25 + 13	+ 25 + 13	+ 38 + 13
		Non-adj.	+ 23 + 10	+ 25 + 13	+ 25 + 13	+ 38 + 13
	Low speed, heavy duty, continuous service, shock, etc.	Adjust.	—	+ 45 + 20	+ 50 + 25	+ 55 + 30
		Non-adj.	—	+ 45 + 20	+ 50 + 25	+ 55 + 30

TABLE 46

Timken Tapered Roller Bearing Housing Fits: Metric I.S.A. Bearings

(Unit = 0·001 mm)

Industry	Type of Application	Cup	Bearing Outside Dia., mm			
			-50	50·1–80	80·1–120	120·1 –180
Automotive	Front wheels, fully floating rear wheels, pinion, differential.	Non-adj.	− 25 − 38	− 28 − 53	− 50 − 75	− 50 − 88
	Rear wheels (private cars), transmission, cross shaft, other applications.	Adjust.	− 0 − 13	− 0 − 25	− 0 − 25	− 0 − 38
Industrial	Machine tool spindles, precision bearings.	Adjust. Non-adj. Floating	* − 25 − 38 *	* − 25 − 38 *	* − 25 − 38 *	* − 25 − 50 *
	Transmission, speed reducers, electrical machinery, sheaves, crown blocks, low speed, heavy duty, continuous service, shock loads.	Adjust. Non-adj. Floating	− 0 − 15 − 25 − 38 + 25 + 10	− 0 − 25 − 28 − 53 + 35 + 10	− 0 − 25 − 50 − 75 + 30 + 5	− 0 − 38 − 50 − 88 + 25 + 0
	Tractor front wheels, implement wheels, etc.	Non-adj.	− 25 − 50	− 25 − 50	− 50 − 75	− 50 − 88

* By selection.

TABLE 47

Bantam Needle Bearing Recommendations

Shaft Dia., in.	Max. Interference, in.	Housing Dia., in.	Max. Interference, in.
0·75–2	0·0010	1·25 – 3·25	0·0007
2·01–3	0·0012	3·26 – 4·5	0·0013
3·01–4·25	0·0016	4·51 – 6·5	0·0017
4·26–6	0·0018	6·51 – 9·125	0·0020
6·01–8	0·0020	9·126 -11·125	0·0025

TABLE 48

Needle Bearing Clearances

Pitch Circle Diameter		Diametral Clearance			
		Minimum		Maximum	
in.	mm	in.	mm	in.	mm
-2·5	0– 60	0·0008	0·020	0·002	0·050
2·501–4	60·1–100	0·001	0·025	0·0025	0·063
4·001–5·125	100·1–130	0·0012	0·030	0·0025	0·063
5·126–6	130·1–150	0·0014	0·035	0·003	0·076
6·001–8	150·1–200	0·0016	0·040	0·003	0·076

MISCELLANEOUS REFERENCES AND TABLES

1. Oil Seal Tolerances

(*a*) "Perfect."

For internal type of seal, sealing on shaft.

(A) Recommended Housing Limits: $\pm$ 0·001 in.; $\pm$ 0·025 mm

(B) Limits on Oil Seal Assembly—

Outer diameter:	$\pm$ 0·002 in.;	$\pm$ 0·05 mm
Width:	+ 0·005 in.;	+ 0·125 mm
	— 0·010 in.;	— 0·25 mm

(*b*) B.S. 1399/1947.

	Housing Dia., in.	*Limits, in.*
Recommended Housing Limits:	0·75 –2·0	+ 0·001, + 0
	2·125–4·0	+ 0·0015, + 0
	4·125–6·0	+ 0·002, + 0
	6·125–8·0	+ 0·003, + 0

2. Tolerances Recommended for Furnace Copper Brazing

Good practice in the use of this process to assemble parts together properly, particularly where airtight results are required, is to have a light interference class of fit. To prevent excessive interference, however, affecting the flow of copper, the use of I.S.A. fit H7/j6 is recommended in conjunction with selective assembly. For further details, see *Aircraft Production*, September, 1943.

3. Tolerances on Drilled Holes

Practice in different firms varies considerably on this question, but Table 49 can be considered as a reasonable schedule of tolerances for drilled holes.

TABLE 49

Diameter, in.	Unit = 0·001 in.	
	High Quality Practice	Normal Practice
–0·04	+ 1, – 0·5	+ 2, – 1
0·041–0·125	+ 1·5, – 0·5	+ 2·5, – 1·5
0·126–0·25	+ 2·5, – 1	+ 4, – 2
0·251–0·5	+ 4, – 1	+ 6, – 2
0·501–0·75	+ 5, – 2	+ 8, – 2
0·751–1	+ 8, – 2	+ 12, – 3
1·01 –2	+ 10, – 2	+ 16, – 4

4. Self-lubricating (Sintered) Bearings

Details of manufacturing accuracy of sintered bushes are given in Chapter VIII. Makers' recommendations for fitting interference are shown in Table 50.

TABLE 50

PRESS FITS FOR OILITE BEARINGS

(Dimensions in Inches)

Nominal o/d of Bearing	Press Fit	
	Min.	Max.
To 0·50	0·0008	0·0018
0·75	0·0011	0·0024
1·00	0·0015	0·0028
1·25	0·0017	0·0032
1·50	0·0020	0·0035
1·75	0·0022	0·0042
2·00	0·0025	0·0045
2·25	0·0027	0·0052
2·50	0·0030	0·0055
2·75	0·0034	0·0065
3·00	0·0038	0·0075

The interferences recommended for "Compo" bearings vary between 0·0025 and 0·004 in., rather unsystematically.

5. Press Fits

Built-up crankshafts of the type used on railway locomotives or Diesel engines where the journal shafts and webs are pressed together and rely on friction only to transmit the torque are common, as manufacture is much easier than with a solid forged shaft. Considerable interference is required to enable the loads to be transmitted, and press loads of 10–15 tons per inch of diameter are usual. The following information is quoted in a paper* at the Institution of Mechanical Engineers as being L.M.S. practice—

Shaft Diameter, in.	*Interference, in.*
4·5	0·009
7	0·015
9	0·015

These fits correspond approximately with I.S.A. "x" shafts (H7–x6).

6. Hexagon Tolerances

British standard bolt and nut hexagon tolerances are given in Table 51. American standard equivalents are given in Table 52.

* *Proc. Inst. Mech. Eng.*, Vol. 127, 1934, p. 249.

TABLE 51

BRITISH STANDARD BOLT AND NUT HEXAGON TOLERANCES

Nominal Hex. Across Flats	Bolts			
	Black (B.S. 916)	Commercial (B.S. 1083)	Aircraft and Standard Whit. (B.S. 190)	B.A. (B.S. 57)
0·087			—	0·003
0·117			0·002	0·003
0·152			0·002	0·003
0·193			0·003	0·004
0·248			0·003	0·004
0·282			—	—
0·324			0·003	0·004
0·365			0·003	—
0·413			0·003	0·004
0·445	0·010	0·007	0·005	
0·525	0·010	0·007	0·005	
0·600	0·015	0·008	0·005	
0·710	0·015	0·008	0·005	
0·820	0·020	0·008	0·005	
0·920	0·020	0·008	0·005	
1·010	0·025	0·010	0·008	
1·100	—	—	0·008	
1·200	0·025	0·010	0·008	
1·300	0·030	0·012	0·008	
1·390	—	—	0·008	
1·480	0·030	0·012	0·012	
1·670	0·030	0·012	0·012	
1·860	0·045	0·015	0·015	
2·050	0·045	0·015	0·015	
2·220	0·045	0·020	0·020	
2·410	0·045	—	0·020	
2·580	0·060	0·025	0·025	
2·760	0·060	0·025	0·025	

TABLE 52

American Standard Hexagon and Square Tolerances

[These tolerances are generally suitable for cold heading as well as rolling or machining of bolts or nuts]

Nominal Hex. or Square A/F	Regular		Heavy Bolts and Nuts	Cap Screw Bolts	Light Nuts	Machine Screws and Nuts
	Bolts	Nuts				
0·1562						0·0062
0·1875						0·0075
0·250						0·009
0·3125						0·0105
0·3437						0·0117
0·375	0·013					0·013
0·4375	—	0·0125		0·0095	0·0095	0·0145
0·500	0·016	—	0·012	0·011	0·011	—
0·5625	0·0185	0·0155	—	0·0115	0·0115	0·0175
0·5938	—	—	0·0158	—	—	—
0·625	0·022	0·019	—	0·013	0·013	0·018
0·6875	—	—	0·0185	—	—	
0·750	0·025	0·022	—	0·013	0·014	
0·7812	—	—	0·0222	—	—	
0·8125	—	0·0245	—	0·0145	—	
0·875	0·028	0·028	0·025	0·015	0·014	
0·9375	0·0315	—	0·0285	—	0·0155	
1·00	—	0·031	—	0·017	—	
1·0625	—	—	0·0315	—	0·0175	
1·125	0·037		—	0·019	—	
1·25	—		0·038	—	0·019	
1·3125	0·0435		—	0·0205	—	
1·4375	—		0·0435	—	0·0205	
1·500	0·050		—	0·023	—	
1·625	—		0·050	—	0·023	
1·6875	0·0565		—	0·0245	—	
1·8125	—		0·0565		0·0245	
1·875	0·063		—		—	
2·00	—		0·062		0·027	
2·0625	0·0685		—		—	
2·1875	—		0·0685		0·0285	
2·25	0·075		—			
2·375	—		0·075			
2·4375	0·0815		—			
2·5625	—		0·0815			
2·625	0·087		—			
2·75	—		0·086			
2·8125	0·0935		—			
2·9375	—		0·0935			
3·00	0·100		—			
3·125	—		0·100			
3·375	0·113		—			
3·50	—		0·112			
3·75	0·125		—			
3·875	—		0·125			
4·125	0·137		—			
4·25	—		0·138			
4·50	0·150		—			
4·625	—		0·150			

Note. The above tolerances for regular and heavy bolts and nuts are derived from the formula—

Tolerance $= 0{\cdot}050D$ where D is the bolt diameter,

or approx. $= 0{\cdot}033W$ where W is the width across flats.

(For cap screws and light nuts, Tolerance $= 0{\cdot}015D + 0{\cdot}006$.)

PART III: MISCELLANEOUS PRODUCTION PROBLEMS

CHAPTER VI

TOLERANCES IN MISCELLANEOUS FABRICATION PROCESSES

ROLLING

Sources of Error

Errors in rolling of sheet or section are—

On average thickness.
On thickness at any point (i.e. flatness).
On section form.

Errors in rolling bar are—

On section form.

Comments on these Errors

Errors on thickness are due to setting errors on the rolls themselves, especially lack of parallelism; distortion of the rolls, especially at the centre of a long roll; eccentricity of the rolls; and naturally wear of the rolls themselves.

Errors on rolled section are firstly in the section form as reproduced in the rolls themselves; secondly, across the section due to roll distortion, gap setting errors, etc.; and finally due to the two rolls not matching properly axially.

Errors in rolled bar are mainly in the rolls themselves or are due to wear.

Representative Limits

Tolerances on rolled bars are given in Tables 53, 54, 55, and 56.

Tolerances on rolled sheet and strip are given in Tables 57, 58, 9, 60, and 61.

TABLE 53

TOLERANCES ON ROLLED BAR: STEEL

(B.S. 32, etc.)

Dia. or Width across Flats, in.	Round	Rectangular
	Tolerance: − 0 to −, in.	
0·125–0·1875	0·002	—
0·188–0·5	0·003	0·003
0·501–0·875	0·003	0·004
0·876–2	0·004	0·005
2·01 –3	0·005	0·006
3·01 –4	0·005	0·006
4·01 –5	0·006	0·006
5·01 –	0·007	0·007

TABLE 54

TOLERANCES ON ROLLED BAR: BRASS

(B.S. 249, 250, etc.)

Dia. or Width across Flats, in.	Round or Rectangular Tolerance: + 0 to +, in.
0–0·875	0·010
0·876–1·25	0·015
1·26 –1·75	0·020
1·76 –2	0·025
2·01 –3	1·5%
3·01 –4	2%
4·01 –	3%

TABLE 55

Tolerances on Rolled Bar: Light Alloy

B.S. 477, L.1, etc.

Dia. or Width across Flats, in.	Round or Rectangular ± ½ Tolerance, in.
–0·4375	0·004
0·438–0·875	0·006
0·876–1·375	0·008
1·376–1·75	0·010
1·76 –2·0	0·012
2·01 –2·625	0·016
2·626–3·0	0·020
3·01 –	2%

***S.A.E.* A.M.S. 2201A**

Dia., in.	Rounds-Rolled
1·501–2·0	± 0·006
2·001–3·499	± 0·008
3·500–5·0	+ 0·031/– 0·016
5·001–8·0	+ 0·063/– 0·031

Width across Flats, in.	Hexagons, Squares, Rectangles, etc.—Rolled
–0·5	± 0·006
0·501–0·75	± 0·008
0·751–1·0	± 0·012
1·001–2·0	± 0·016
2·001–3·0	± 0·020

Dia., Width across Flats, etc., in.	Cold Finished		
	Rounds	Squares, Hex., etc.	Rect-angles
–0·035	± 0·0005	—	—
0·036–0·064	± 0·001	± 0·0015	± 0·0015
0·065–0·5	± 0·0015	± 0·002	± 0·002
0·501–1·0	± 0·002	± 0·0025	± 0·0025
1·001–1·5	± 0·0025	± 0·003	± 0·003
1·501–2·0	± 0·004	± 0·005	± 0·005
2·001–3·0	± 0·004	—	± 0·005
3·001–4·0	—	—	± 0·005

TABLE 56

BRITISH STANDARD TOLERANCES FOR HEXAGON BARS

(All Tolerances: — 0 to —, in.)

Nominal Hex. across Flats, in.	Steel	Non-ferrous	
		Extruded or Rolled	Drawn after Rolling
0·087	—	0·003	0·002
0·117	—	0·003	0·002
0·152	—	0·003	0·002
0·193	0·003	0·004	0·002
0·248	0·003	0·004	0·002
0·282	0·003	0·004	0·002
0·324	0·003	0·004	0·002
0·365	0·003	0·004	0·002
0·413	0·003	0·004	0·002
0·445	0·003	0·004	0·002
0·525	0·003	0·004	0·002
0·600	0·004	0·004	0·003
0·710	0·004	0·004	0·003
0·820	0·004	0·005	0·003
0·920	0·004	0·005	0·003
1·010	0·005	0·005	0·004
1·100	0·005	0·005	0·004
1·200	0·005	0·005	0·004
1·300	0·005	0·005	0·004
1·390	0·005	—	—
1·480	0·005	0·006	0·004
1·670	0·005	0·006	0·004
1·860	0·005	0·006	0·004
2·050	0·005	0·008	0·004
2·220	0·006	0·008	—
2·410	0·006	0·008	—
2·580	0·006	0·010	—
2·760	0·006	0·010	—

TABLE 57

Tolerances on Sheet Steel—Aircraft (S.3, D.T.D.124A, etc.), Cold Rolled

Sheet Gauge, in.	Width, in.									
	0–3·99		4–5·99		6–9·99		10–15·99		16–24	
	Centre $\frac{3}{4}$ + 0 to +	Outer $\frac{1}{8}$	Centre $\frac{3}{4}$ + 0 to +	Outer $\frac{1}{8}$	Centre $\frac{3}{4}$ + 0 to +	Outer $\frac{1}{8}$	Centre $\frac{3}{4}$ + 0 to +	Outer $\frac{1}{8}$	Centre $\frac{3}{4}$ + 0 to +	Outer $\frac{1}{8}$
–0·019	1	+ 1 − 0·5	1	+ 1 − 0·5	1·5	+ 1·5 − 1	1·5	+ 1·5 − 1	2	+ 2 − 1·5
0·020–0·031	1·5	+ 1·5 − 0·5	1·5	+ 1·5 − 1	2	+ 2 − 1	2·5	+ 2·5 − 1	3	+ 3 − 1·5
0·032–0·047	2	+ 2 − 0·5	2	+ 2 − 1	2	+ 2 − 1·5	2·5	+ 2·5 − 1·5	3·5	+ 3·5 − 1·5
0·048–0·063	2·5	+ 2·5 − 1	2·5	+ 2·5 − 1·5	3	+ 3 − 1·5	3·5	+ 3·5 − 1·5	3·5	+ 3·5 − 1·5
0·064–0·091	3	+ 3 − 1	3	+ 3 − 1·5	4	+ 4 − 2	4·5	+ 4·5 − 2	4·5	+ 4·5 − 2
0·092–0·127	3·5	+ 3·5 − 1·5	3·5	+ 3·5 − 2	4·5	+ 4·5 − 2·5	5	+ 5 − 2·5	5	+ 5 − 2·5
0·128–0·159	4	+ 4 − 1·5	4	+ 4 − 2	5	+ 5 − 2·5	5·5	+ 5·5 − 2·5	6	+ 6 − 2·5
0·160–0·191	4·5	+ 4·5 − 1·5	4·5	+ 4·5 − 2	5·5	+ 5·5 − 3	6	+ 6 − 3	6·5	+ 6·5 − 3
0·192–0·232	5	+ 5 − 2	5	+ 5 − 2·5	6	+ 6 − 3	6·5	+ 6·5 − 3	7	+ 7 − 3

Tolerance unit = 0·001 in.

TABLE 58

Tolerances on Sheet Steel—Aircraft (S.3, D.T.D.124A, etc.), Hot Rolled and Sheet, Light Alloy, Aircraft (L.3, etc.)

Sheet Gauge, in.	Tolerance: + 0 to +, in.
–0·027	0·004
0·028–0·047	0·005
0·048–0·091	0·006
0·092–0·143	0·010
0·144–0·191	0·012
0·192–0·232	0·014

TABLE 59

Tolerances on Light Alloy Strip—Aircraft (L.3, etc.)

Sheet Gauge, in.	Width of Strip, in.		
	0–12	12·1–16	16·1–20
	Tolerance: + 0 to +, in.	Tolerance: + 0 to +, in.	Tolerance: + 0 to +, in.
0·008	0·002	—	—
0·012	0·002	—	—
0·016	0·002	0·002	—
0·020, 0·022	0·002	0·003	0·003
0·024–0·028	0·003	0·003	0·003
0·032–0·040	0·004	0·004	0·004
0·041–0·080	0·005	0·005	0·005
0·081–0·104	0·007	0·007	0·007
0·105–0·144	0·008	0·008	0·008
0·145–0·192	0·010	0·010	0·010

TABLE 60

Tolerances on Sheet, Light Alloy: General Engineering (B.S. 395, 414, etc.)

Sheet Gauge, in.	Tolerance, in. (± ½ Tolerance)*
–0·027	0·004
0·028–0·047	0·006
0·048–0·091	0·008
0·092–0·143	0·010
0·144–0·192	0·012
0·193–	+ 10%, + 0

* For sheets over 3 ft wide add + 0·002 in.

TABLE 61

Tolerances on Brass and Bronze Sheet, Strip, and Foil (B.S. 265, 407, etc.)

Sheet Gauge, in.	Tolerances (in.): All Limits ± ½ Tolerance				
	Width of Sheet or Strip, in.				
	0–6	6·1–12	12·1–24	24·1–36	36·1–42
–0·009	0·001	0·002	0·003	—	—
0·010–0·023	0·002	0·003	0·004	0·005	—
0·024–0·039	0·003	0·004	0·005	0·006	0·007
0·040–0·079	0·004	0·005	0·006	0·007	0·008
0·080–0·127	0·004	0·005*	0·007	0·008	0·010
0·128–0·25	0·005	0·006†	0·008	0·010	0·012

* 0·006 for Spec. B.15 and B.16.
† 0·007 for Spec. B.15 and B.16.

STAMPING

1. Steel Stampings and Forgings

Sources of Error

Dimensional errors on stampings, apart from die faults, arise from—

Variation in thickness, measured normally to the die split line.
Die wear.
Malalignment of the die halves.
Shrinkage.

Comments on these Errors

Thickness variation is mainly due to excess metal being put between the dies.

Die wear is due to erosion or abrasion of the die during stamping; this will normally be in the oversize or plus direction, increasing stamping weight, but will be worse at some places than at others.

Malalignment of the two halves of the die is due to the alignment keys or pins, etc., wearing.

Shrinkage of the stamping as it cools should be allowed for in die making, but may cause trouble due to uneven shrinkage. Shrinkage may be up to 0·004 in. per inch along the die line.

Representative Limits

THICKNESS—

Small steel stamping or medium light alloy stamping	+ 0·015 in. — 0·005 in.
Large high-quality aircraft stamping (say 50 lb weight)	+ 0·060 in. — 0·020 in.
Large commercial stamping (say 50–100 lb)	+ 0·125 in. — 0·040 in.
Small stamping cold-coined	± 0·005 in.

MALALIGNMENT—

Maximum limit:	
Small stamping	0·010 in.
Large stamping (say 50 lb steel or equivalent)	0·030 in.

Details of American stamping limits are given in Tables 61, 62, and 63.

2. Cold Heading (*Machine Design*, July, 1946)

On diameters formed in the die: minimum, ± 0·001 in.; normal, ± 0·002 in.; for maximum die life, ± 0·004 in.

On overall length outside the die head, minimum ± 0·015 in.; normal, ± 0·032 in.

3. Light Alloy Stampings

The tables for steel stampings can be worked to for light alloy stampings, taking the equivalent steel stamping weight as three times the true alloy stamping weight.

TABLE 62

AMERICAN DROP FORGING ASSOCIATION LIMITS FOR STEEL STAMPINGS (UP TO 100 LB IN WEIGHT)
Tolerance unit = 0·001 in.

Stamping Weight, lb	Thickness Normal to Die Line	
	Commerical	Close
0·2	+ 24, − 8	+ 12, − 4
0·4	+ 27, − 9	+ 15, − 5
0·6	+ 30, − 10	+ 15, − 5
0·8	+ 33, − 11	+ 18, − 6
1	+ 36, − 12	+ 18, − 6
2	+ 45, − 15	+ 24, − 8
3	+ 51, − 17	+ 27, − 9
4	+ 54, − 18	+ 27, − 9
5	+ 57, − 19	+ 30, − 10
10	+ 66, − 22	+ 33, − 11
20	+ 78, − 26	+ 39, − 13
30	+ 90, − 30	+ 45, − 15
40	+ 102, − 34	+ 51, − 17
50	+ 114, − 38	+ 57, − 19
60	+ 126, − 42	+ 63, − 21
70	+ 138, − 46	+ 69, − 23
80	+ 150, − 50	+ 75, − 25
90	+ 162, − 54	+ 81, − 27
100	+ 174, − 58	+ 87, − 29

TABLE 63

AMERICAN DROP FORGING ASSOCIATION LIMITS FOR STEEL STAMPINGS (UP TO 100 LB IN WEIGHT)
Tolerance unit = 0·001 in.

Limits on Length Parallel to Die Line					
Total =	Shrinkage		+	Die Wear Allowance	
Length, in.	Commercial	Close	Weight, lb	Commercial	Close
1	± 3	± 2	1	± 32	± 16
2	± 6	± 3	3	± 35	± 18
3	± 9	± 5	5	± 38	± 19
4	± 12	± 6	7	± 41	± 21
5	± 15	± 8	9	± 44	± 22
6	± 18	± 9	11	± 47	± 24
For each 1 in.	add ± 3	add ± $1\frac{1}{2}$	For each 2 lb	add ± 3	add ± $1\frac{1}{2}$

TABLE 64

AMERICAN DROP FORGING ASSOCIATION LIMITS FOR STEEL STAMPINGS (UP TO 100 LB IN WEIGHT).

Tolerance unit = 0·001 in.

Die Malalignment		
Stamping Weight, lb	Commercial	Close
1	15	10
7	18	12
13	21	14
19	24	16
25	27	18
31	30	20
For each 6 lb	add 3	add 2

DRAWING

1. Tube Drawing

Sources of Error

Errors on tube drawing or rolling processes are—

On inside diameter.
On outside diameter.
On concentricity of these two (i.e. wall thickness).
On circularity (e.g. ovality).
On coaxiality (i.e. the minimum wall thickness is not along an axial line).
On straightness.

Comments on these Errors

Diameter errors are due to errors in or wear of the drawing dies.

Lack of concentricity is due to initial error in the rough "hollow" which is made from a pierced billet; it can be minimized by using a cold centred billet, or by machining the tube on a mandrel after the first draw. Eccentricity cannot be corrected or lost during drawing itself.

Ovality is due in part to internal stress in the tube and is thus inherent in the process. It can be aggravated by non-uniform die wear. Straightening processes if adopted may cause ovality.

Coaxiality may be lost owing to a combination of other errors. In the case where the position of minimum wall thickness is twisted down the tube, it is due to the way the hot rolled rough tube is fed into the rolls, being turned somewhat at each pass.

Lack of straightness or bow is inherent owing to contraction stress on cooling; it can be corrected by straightening.

Representative Limits

CONCENTRICITY. Good practice with normal billets: within ± 10 per cent of nominal wall thickness.

Best practice with centred billets: within ± 5 per cent of nominal wall thickness.

SPECIFICATION LIMITS. Tolerances on brass and copper tubes are given in Tables 65, 66, and 67.

TABLE 65

LIMITS ON BRASS AND COPPER TUBES—Aircraft (T.7, etc.) —OUTSIDE DIAMETER

Outside Dia., in.	Tolerance, in. (± ½ Tol.)
–0·625	0·006
0·626–2·0	0·008
2·01 –2·5	0·010
2·51 –	1%

(Soft copper tubes (e.g. to B.S. 1401) can be drawn to a diameter tolerance of 0·003 in. up to 2 in. diameter.)

TABLE 66

LIMITS ON BRASS AND COPPER TUBES—WALL THICKNESS—COPPER

Wall Thickness t, in.	Limits		
	Mean t, in.	Max. t, in.	Min. t, in.
0·022	± 0·002	+ 0·003	– 0·003
0·028	± 0·002	+ 0·004	– 0·004
0·036	± 0·003	+ 0·005	– 0·005
0·048	± 0·003	+ 0·006	– 0·006
0·064	± 0·004	+ 0·007	– 0·007

TABLE 67

LIMITS ON BRASS AND COPPER TUBES—WALL THICKNESS—BRASS

Wall Thickness t, in.	Limits		
	Mean t, in.	Max. t, in.	Min. t, in.
0·018–0·022	± 0·002	+ 0·003	– 0·003
0·024–0·036	± 0·003	+ 0·004	– 0·004
0·040–0·072	± 0·004	+ 0·006	– 0·006
0·080–0·116	± 0·006	+ 0·010	– 0·010
0·128–0·192	± 0·008	+ 0·013	– 0·013

*Aircraft Tube Tolerances (T45, etc.)**

Tube is specified by either outside or inside diameter and wall thickness. Limits on diameter refer to the specified diameter.

LIMITS ON MEAN DIAMETER

$$\text{Dia.}_{\text{mean}} = \frac{\text{Dia.}_{\text{max.}} + \text{Dia.}_{\text{mean}}}{2}$$

Limits on $\text{Dia.}_{\text{mean}} = \pm$ 0·003 in. for dia. up to 1·5 in.
$\pm$ 0·001 in. for each 0·5 in. (or part) above 1·5 in.

LIMITS ON EXTREME DIAMETER

(*a*) Drawn and tempered tubes—

$$\text{Dia.}_{\text{max.}} \text{ or } \text{Dia.}_{\text{mean}} = \pm\left(0{\cdot}005 + \frac{0{\cdot}6D^3}{(1000t)^2}\right) \text{in.}$$

or Limits on $\text{Dia.}_{\text{mean}}$ + 0·002 in.

whichever is the greater.

(Fourth decimal place *ignored*, i.e. 0·0019 = 0·001 in.)

(*b*) Hardened and tempered tubes—

$$\text{Dia.}_{\text{max.}} \text{ or } \text{Dia.}_{\text{mean}} = \pm\left(0{\cdot}005 + \frac{D^3}{(1000t)^2}\right)$$

or Limits on $\text{Dia.}_{\text{mean}}$ + 0·002 in.

whichever is the greater.

(Fourth decimal place *ignored*, i.e. 0·0019 = 0·001 in.)

LIMITS ON THICKNESS

Thickness t, in.	Limits on t_{mean}, in.
–0·022	+ 0·003 + 0
0·024–0·056	+ 0·004 + 0
0·062–	+ 8% + 0

Also, $t_{\text{max.}} = 1{\cdot}1 \times t_{\text{nominal}}$ + tolerance on t_{mean}

$t_{\text{min.}} = 0{\cdot}9 \times t_{\text{nominal}}$.

* Information on American aircraft tube tolerances, not greatly different from British practice, is contained in the S.A.E. standard, A.M.S.2253B (1947).

2. Bar, Rod or Wire Drawing

Sources of Error

Errors in rod or wire drawing are due to—

Die wear.
The process itself.

Comments on these Errors

Die wear is an obvious source of error but work of high quality is produced particularly by using multiple drawing operations. There are miscellaneous causes of error in the process itself, temperature variations, speed of drawing variations, etc. These variations are not, however, serious, as can be appreciated when the accuracy obtained commercially is considered.

Representative Limits

Tolerances for drawn bar and wire are given in the Tables 67–71.

TABLE 68

BRITISH STANDARD TOLERANCES FOR BRIGHT DRAWN BAR

Dia. or Width across Flats, in.	Tolerance: – 0 to –, in.	
	Round	Rectangular
0·125–0·1875	0·002	—
0·188–0·5	0·003	0·003
0·501–0·875	0·003	0·004
0·876–2	0·004	0·005
2·01 –3	0·005	0·006
3·01 –4	0·005	0·006
4·01 –5	0·006	0·006
5·01 –	0·007	0·007

TABLE 69

NON-FERROUS BARS DRAWN AFTER ROLLING OR EXTRUDING (B.S. 249, 250, etc.)

Dia. or Width across Flats, in.	Tolerance: – 0 to –, in.
–0·1875	0·0015
0·188–0·75	0·002
0·751–1·375	0·003
1·376–2	0·004
2·01 –3	± 0·5%*
3·01 –	± 1%*

* These seem excessive.

TABLE 70

LIMITS ON STEEL WIRE (W.1)

Wire Gauge, in.	Tolerance: + 0 to +, in.
0·022–0·024	0·001
0·028–0·072	0·002
0·080–0·128	0·003
0·144–0·160	0·004

TABLE 71

M.O.S. S.T.A. SPRING WIRE LIMITS

Wire Gauge, in.	Limits on Dia., in.	Tolerance on Ovality, in.
0·010–0·064	± 0·0005	0·0005
0·065–	± 0·001	0·001

TABLE 72

NORMAL BRITISH WIRE DRAWER'S LIMITS (B.S. 1408)

Wire Gauge, in.	Limits, in.
0·0078–0·018	± 0·00025
0·019 –0·064	± 0·0005
0·065 –0·192	± 0·001
0·193 –	± 0·0015

EXTRUSION

Sources of Error

Errors with the extrusion process are due to—

Unbalanced flow of metal through die or unsymmetrical sections.
Die wear.
Speed of extrusion.
Heat treatment.

Comments on these Errors

Press design and its feeding arrangements, or unsymmetrical sections cause variations in speed of flow during extrusion and only properly balanced sections can be made reliably; unsymmetrical sections may need straightening after extrusion.

Die wear is related to the shape of the section, unsupported or awkward parts of the die may wear more rapidly than simple parts; distortion of parts of the die may occur.

Speed of extrusion affects the final shape, since it affects the strain in the material during extrusion.

Heat treatment causes distortion on long lengths and particularly on hollow sections (such as a "U") of appreciable depth.

Representative Limits

See Table 73.

TABLE 73

TOLERANCES ON EXTRUDED BAR: NON-FERROUS MATERIALS (Brass, Duralumin (B.S. 477), etc.)

Dia. or Width across Flats, in.	Tolerance, in. ($\pm \frac{1}{2}$ Tolerance)
–0·4375	0·004
0·438–0·875	0·006
0·876–1·375	0·008
1·376–1·75	0·010
1·76 –2	0·012
2·01 –2·625	0·016*
2·626–3	0·020*
3·01 –	2%

TOLERANCES FOR SPECIAL SECTIONS—AIRCRAFT LIGHT ALLOYS (L1, L40, ETC.)

(*a*) *Rectangular Section Bar*

Tolerance on width $W = \pm (0{\cdot}007 + 0{\cdot}006W)$ in.

Tolerance on thickness $t = \pm (0{\cdot}007 + 0{\cdot}003(W + t))$ in.

(*b*) *Special Sections* (*Angles, Tees, etc.*) (except as indicated below in (*c*))

Tolerance on overall width $W = \pm (0{\cdot}007 + 0{\cdot}006W)$ in.

Tolerance on thickness t ($W < 3$ in.)—see Table 74.

Tolerance on thickness

($W = 3{\cdot}01 - 4{\cdot}0$ in.) = Table 74 Tolerance $\times$ 1·33 (Max. $\pm$ 0·05)

($W = 4{\cdot}01 - 5{\cdot}0$ in.) = Table 74 Tolerance $\times$ 1·66 (Max. $\pm$ 0·05)

Tolerance on all angles: $\pm 2°$.

* Quoted as 1% in some specifications.

(*c*) *Deep Channels*

(Nominal depth d/nominal width $W > 1$)

Tolerance on height or width of base (not subject to spring)

$$= \pm (0{\cdot}007 + 0{\cdot}006W)$$

Tolerance on overall dimensions, internal and external, across mouth of section

$$d/W = 1{\cdot}01\text{–}2 : \pm 2(0{\cdot}007 + 0{\cdot}006W)$$
$$d/W = 2{\cdot}01\text{–}3 : \pm 2{\cdot}5(0{\cdot}007 + 0{\cdot}006W)$$
$$d/W = 3{\cdot}01\text{–} \quad : \pm 3(0{\cdot}007 + 0{\cdot}006W)$$

Tolerance on thickness t: as Table 74.

TABLE 74

TOLERANCE ON THICKNESS OF SPECIAL LIGHT ALLOY SECTIONS
($W \nless 3$ in.)

Thickness t, in.	Tolerance, in.
–0·08	± 0·008
0·081–0·125	± 0·009
0·126–0·187	± 0·010
0·188–0·250	± 0·012
0·251–0·375	± 0·015
0·376–0·500	± 0·018
0·501–0·625	± 0·022
0·626–	± 3% (Max. ± 0·05)

(*d*) *Tapered Sections*

The tolerance on thickness at any point shall be that given in Table 74 for the particular nominal thickness involved.

TOLERANCES FOR ALUMINIUM ALLOY EXTRUDED SECTIONS (as published by the Wrought Light Alloy Association)

(*a*) *Angles*

Sections, rectangular bars and similar shapes (thickness t)

$$t < 0{\cdot}2 \text{ in.—tolerance} = \pm 2^\circ$$
$$t > 0{\cdot}2 \text{ in.—tolerance} = \pm 1\tfrac{1}{2}^\circ$$

If the section includes an angle with two thicknesses, use the smaller for t.

(*b*) *Regular Section Bars*

As Table 73 above.

(*c*) *Tolerances on Overall Widths of Regular Sections*

See Table 75.

TABLE 75

TOLERANCES ON OVERALL WIDTHS OF REGULAR SECTIONS

Width W, in.	Tolerance on W, in. ($\pm \frac{1}{2}$ Tolerance)
0–0·149	0·014
0·15–0·199	0·016
0·2 –0·299	0·016
0·3 –0·399	0·018
0·4 –0·499	0·020
0·5 –0·749	0·022
0·75–0·999	0·024
1 –1·499	0·030
1·5 –1·999	0·036
2 –2·499	0·042
2·5 –2·999	0·048
3 –3·999	0·054
4 –4·999	0·060
5 –5·999	0·070
6 –6·999	0·080
7 –8	0·090

(*d*) *Tolerances on Thickness of Regular Sections*

See Table 76.

(*e*) *Tolerances for the Open Ends of Right-angled Channels, I-beams, H-sections, etc.*

See Table 77.

Tolerances on Impact Extrusion (*Machine Design*, February, 1947, "Production Processes")

Wall thickness and diameters can be held to $\pm$ 0·001 to $\pm$ 0·002 in.; bottom thicknesses to $\pm$ 0·003 to $\pm$ 0·005 in., depending on complexity.

CASTING

Sources of Errors

Dimensional errors on castings apart from pattern or die faults may be due to—

Core shift.
Faulty moulds especially in sand.
Malalignment of the mould or die sections.
Shrinkage or contraction.
Mould distortion during casting.
Fettling.
Poor design, e.g. unsymmetrical sections.

TABLE 76

Tolerances on Thickness for Regular Sections

Nominal Thickness of Section, in.	For Widths (in.)										
	Up to 0·499	0·500–0·999	1·000–1·499	1·500–1·999	2·000–2·499	2·500–2·999	3·000–3·999	4·000–4·999	5·000–5·999	6·000–6·999	7·000–8·000
0·031–0·049	*	*	*	*	*	*	*	*	*	*	*
0·050–0·074	± 0·007	± 0·008	± 0·008	*	*	*	*	*	*	*	*
0·075–0·099	± 0·007	± 0·008	± 0·008	± 0·009	*	*	*	*	*	*	*
0·100–0·149	± 0·007	± 0·008	± 0·009	± 0·010	± 0·011	*	*	*	*	*	*
0·150–0·199	± 0·008	± 0·009	± 0·010	± 0·011	± 0·012	± 0·013	*	*	*	*	*
0·200–0·299	± 0·008	± 0·009	± 0·010	± 0·011	± 0·012	± 0·013	± 0·014	*	*	*	*
0·300–0·399	± 0·009	± 0·010	± 0·011	± 0·012	± 0·013	± 0·014	± 0·015	± 0·017	*	*	*
0·400–0·499	± 0·010	± 0·011	± 0·012	± 0·013	± 0·014	± 0·015	± 0·016	± 0·018	± 0·019	*	*
0·500–0·749	± 0·011	± 0·012	± 0·013	± 0·014	± 0·015	± 0·016	± 0·017	± 0·019	± 0·020	± 0·022	*
0·750–0·999		± 0·012	± 0·014	± 0·015	± 0·016	± 0·017	± 0·018	± 0·020	± 0·021	± 0·024	± 0·027
1·000–1·499			± 0·015	± 0·016	± 0·017	± 0·018	± 0·019	± 0·021	± 0·023	± 0·026	± 0·029
1·500–1·999				± 0·018	± 0·019	± 0·020	± 0·021	± 0·022	± 0·024	± 0·028	± 0·031
2·000–2·499					± 0·021	± 0·022	± 0·023	± 0·024	± 0·026	± 0·030	± 0·033
2·500–2·999						± 0·024	± 0·025	± 0·026	± 0·028	± 0·032	± 0·035
3·000–3·999							± 0·027	± 0·028	± 0·029	± 0·034	± 0·037
4·000–4·999								± 0·030	± 0·031	± 0·036	± 0·039
5·000–5·999									± 0·035	± 0·038	± 0·041
6·000–6·999										± 0·040	± 0·043
7·000–8·000											± 0·045

* To be regarded as special sections.

TABLE 77

Tolerances for the Open Ends of Right-angled Channels, I-beams, H-sections, etc.

Overall Width of Channel, in.	Minimum Thickness of Base or Leg, in.	External or Internal Width Tolerances (in.) at Top of Gap for Depths of 0·500–0·999	1·000–1·499	1·500–1·999	2·000–2·499	2·500–2·999	3·000–3·499	3·500–3·999	4·000–4·499	4·500–4·999	5·000–5·499	5·500–6·000
0·500–0·999	0·050–0·099	± 0·021										
	0·100–0·199	± 0·020	± 0·024									
	0·200–0·299	± 0·019	± 0·023									
	0·300 and over	± 0·018	± 0·021									
1·000–1·499	0·050–0·099	± 0·025	± 0·030									
	0·100–0·199	± 0·024	± 0·028	± 0·033								
	0·200–0·299	± 0·023	± 0·027	± 0·031	± 0·036							
	0·300 and over	± 0·022	± 0·025	± 0·029	± 0·034	± 0·036						
1·500–1·999	0·100–0·199	± 0·027	± 0·031	± 0·036								
	0·200–0·299	± 0·026	± 0·031	± 0·034	± 0·038	± 0·042						
	0·300 and over	± 0·025	± 0·028	± 0·032	± 0·035	± 0·039						
2·000–2·499	0·100–0·199	± 0·030	± 0·034	± 0·039								
	0·200–0·299	± 0·029	± 0·033	± 0·037	± 0·041	± 0·045						
	0·300 and over	± 0·028	± 0·031	± 0·035	± 0·038	± 0·042						
2·500–2·999	0·150–0·199	± 0·033	± 0·037	± 0·042	± 0·046							
	0·200–0·299	± 0·032	± 0·036	± 0·040	± 0·044	± 0·048	± 0·052					
	0·300 and over	± 0·031	± 0·034	± 0·038	± 0·041	± 0·045	± 0·048					
3·000–3·999	0·200–0·299	± 0·035	± 0·039	± 0·043	± 0·047	± 0·051	± 0·055					
	0·300 and over	± 0·034	± 0·037	± 0·041	± 0·044	± 0·048	± 0·051	± 0·055				
4·000–4·999	0·200–0·299	± 0·038	± 0·042	± 0·046	± 0·050	± 0·054						
	0·300 and over	± 0·037	± 0·040	± 0·044	± 0·047	± 0·051	± 0·054					
5·000–5·999	0·300 and over	± 0·040	± 0·043	± 0·047	± 0·050	± 0·054	± 0·057	± 0·061				
6·000–6·999	0·500 and over	± 0·043	± 0·045	± 0·048	± 0·050	± 0·053	± 0·055	± 0·058	± 0·060	± 0·063	± 0·065	± 0·068
7·000–8·000	0·500–0·599	± 0·048	± 0·050	± 0·053	± 0·055	± 0·058	± 0·060	± 0·063	± 0·065	± 0·068	± 0·070	± 0·073
	0·600 and over	± 0·047	± 0·049	± 0·051	± 0·053	± 0·055	± 0·057	± 0·059	± 0·061	± 0·063	± 0·065	± 0·067

Comments on these Errors

Core shift on sand castings is a very common fault, but usually means a decrease of one wall thickness and a corresponding increase of an opposite wall.

Imperfectly made moulds result in errors of the parts of the casting cast in them. This is particularly common on thin sections.

Malalignment of mould or die sections is due to worn or oversize locating dowels or keys.

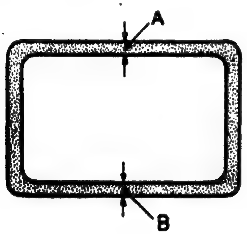
Shrinkage or contraction distortion errors, although allowed for in pattern design, may give trouble locally, particularly with flimsy castings.

Mould distortion may occur during the pouring of large castings.

Fettling and the cutting off of runners, risers, etc., inevitably causes local error at parts of the surface.

Representative Limits

In setting limits of core shift, it should be borne in mind that a reduction of one wall with an increase of the opposite wall is



Tolerance on A or B, $+x$, $-y$.

Either wall may be undersize by y provided the opposite wall is a similar amount oversize.

Therefore tolerance on $A + B = {}^{+2x}_{+0}$

FIG. 80. TOLERANCES ON CASTING WALL THICKNESS

usually not so important as a reduction of both. In this case, the procedure of Fig. 80 may be adopted, provided the wall thicknesses can readily be measured. The tolerance itself depends on the size, weight, and accuracy of location of the core.

TABLE 78

NORMAL TOLERANCE ON SAND CASTING WALL THICKNESS

Section, in.	Light Alloy	Ferrous or Bronze	
		High Quality	Commercial
–0·3	0·03	0·04	0·05
0·31–0·6	0·05	0·06	0·08
0·61–1·0	0·08	0·09	0·12
1·01–	0·10	0·12	0·15

TABLE 79

TOLERANCE ON SAND CASTING THICKNESS ACROSS PATTERN SPLIT PLANE

Thickness, in.	Light Alloy	Ferrous and Bronze	
		High Quality	Commercial
–0·6	0·06	0·06	0·08
0·61–1·5	0·10	0·10	0·12
1·51–	0·15	0·15	0·18

TABLE 80

TOLERANCE ON LENGTH DIMENSION IN A SAND CASTING

Length, in.	High Quality	Commercial
– 0·6	0·03	0·05
6·1–15	0·06	0·10
15·1–40	0·12	0·18
40·1–	0·18	0·25

*Tolerances on Die Castings**

(Note. Unless otherwise stated, these refer to pressure castings in aluminium, magnesium, and zinc.)

1. CENTRES OF HOLES CORED AT RIGHT ANGLES TO PARTING LINE (refer to dimensions a and b, Fig. 81).

$\pm$ 0·002 in. up to 1·5 in., add $\pm$ 0·001 in. for each additional inch or part thereof, e.g. when b = 3·25 in. tolerance will be $\pm$ 0·004 in.

Note. The above is only applicable to centres of holes when cores are located in same die member; when in different members add a further $\pm$ 0·003 in., e.g. when a = 4·5 in. then tolerance will be ($\pm$ 0·005) + ($\pm$ 0·003) = $\pm$ 0·008 in.

2. CENTRES OF CORED HOLES CORED PARALLEL TO PARTING LINE (refer to dimensions k and l).

0·003 in. at dimension k when core C is located as illustrated in die member B.

+ 0·006 in./– 0·004 in. at dimension l when core D is located as illustrated in die member A.

3. ECCENTRICITY TOLERANCES.

Between diameters e and f = 0·001 in. when core is located at both ends as illustrated at F.

Between diameters d, e, f, to c = 0·004 in. when cores E and F are located to each other.

* From the *Journal of the Institution of Production Engineers*, Jan. 1945, Vol. XXIV, No 1.

Note. Above eccentricity tolerances apply only when diameters are 0·375 in. minimum and not more than 1·5 diameters long.

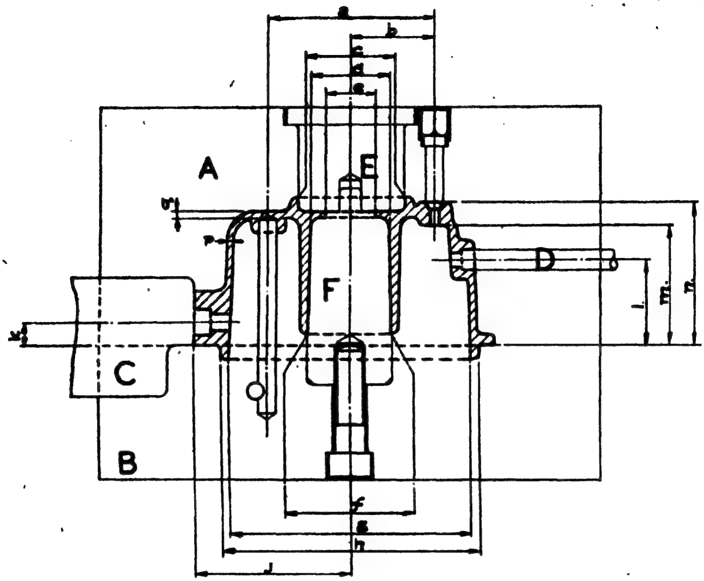


Fig. 81. Diagram Illustrating Section through Typical Pressure Die Casting

Between diameters f, g, $h = 0{\cdot}001$ in. when formed in the same die block as illustrated.

See also Fig. 82.

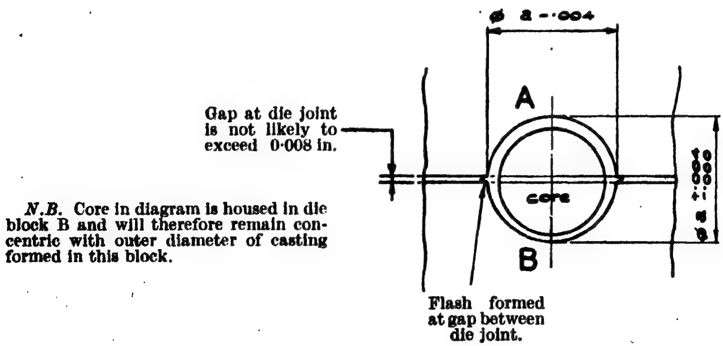


Fig. 82. Diagram Illustrating Typical Effect of Imperfect Die Closure upon the Tolerances of a Cylindrical Casting Produced from an Axially Jointed Die

4. TOLERANCES ON CORED HOLES AND MALE SPIGOTS, ETC.

$\pm$ 0·0015 in. on diameters c, d, e, f, g, h, when not in excess of 0·75 in. diameter, add $\pm$ 0·0015 in. for each additional inch or part thereof; e.g. when $g = 5 \cdot 5$ in. diameter, tolerance will be $\pm$ 0·009 in.

The above refer to aluminium and magnesium; if in zinc base use:

$\pm$ 0·001 in. on diameters c, d, e, f, g, h, when not in excess of 0·75 in. diameter, add $\pm$ 0·001 in. for each additional inch or part thereof.

5. TOLERANCES APPLICABLE TO BOSS FACINGS, ETC.

$\pm$ 0·002 in. up to 1·25 in. at dimension m where both faces are formed in same die member.

Add $\pm$ 0·001 in. for each additional inch or part thereof.

$\pm$ 0·006 in. up to 1·25 in. at dimension n where faces are formed in different members.

Add $\pm$ 0·001 in. for each additional inch or part thereof.

$\pm$ 0·006 in. up to 2 in. at dimension j where boss face is formed by moving core C.

Add $\pm$ 0·0015 in. for each additional inch or part thereof.

6. TOLERANCES OF WALL THICKNESSES.

$\pm$ 0·003 in. at dimension p.

$\pm$ 0·004 in. at dimension q.

MOULDING

1. Plastic Moulding*

Sources of Error

Sources of error in the moulding of thermosetting are—

- In the tools themselves.
- Due to tool wear.
- Due to variations in powder shrinkage.
- Miscellaneous minor variable more or less important depending on the particular process.

Comments on these Errors

ERRORS IN TOOLMAKING. The accuracy of tools normally obtained in precision engineering is of a high order, but it is limited by certain conditions, and during the hardening process distortion may occur. Correction of inaccuracies by grinding after hardening is not always possible owing to intricate cavity shapes.

TOOL WEAR. Tool wear is affected by the number of mouldings produced, the quality of the tool material, the pressures used in

* Most of the information contained in this section is drawn from the British Plastics Federation publications referred to in the Bibliography.

moulding, and the type of filler in the powder. The flow of the powder under pressure has an abrasive effect on the tool; this is the greatest with shock-resisting, hard-flowing powders, or hard mineral fillers. This abrasive effect may be counteracted in special cases by designing important tool parts subjected to wear so that they are interchangeable or replaceable.

POWDER SHRINKAGE. Variation in shrinkage occurs between different batches of the same grade of powder and within each powder group. The variation in powder shrinkage is greatest with wood-filled powders, less with cellulose-filled shock-resisting powders, and least with mineral-filled heat-resisting powders. This difference of shrinkage between batches of the same grade of powder may even exceed 0·003 in. per inch.

GENERAL MOULDING CONDITIONS. All tools consist of two or more parts and a minimum clearance must be allowed between the parts. The clearance will influence the accuracy of the moulding while the clearance itself is affected by the powder filler used; a coarse filler results in thicker flash between the tool parts. With multiple cavity tools, owing to production conditions and misalignment in the tool and the press, greater error may occur than with single cavity tools which can be controlled more precisely. General improvement, with corresponding lower error, may be obtained by using the transfer method of moulding or the high frequency heating technique. When transfer methods are used the flash between the tool parts does not require special consideration. Expansion of the tool, chemical and thermal action and reaction during the moulding processes, and the shrinkage after moulding are not always controllable.

Representative Limits

A system of tolerances for plastic moulding has been published by the British Plastics Federation in their publication No. 5 (April, 1945). This has been proposed for extended trial prior to the issue of a B.S.I. standard.

The derivation of the tolerances is as follows—

$$\begin{aligned} &\text{Fine:} && \pm\, 0{\cdot}5\,(0{\cdot}004D + k)\\ &\text{Medium:} && \pm\, 1{\cdot}0\,(0{\cdot}004D + k)\\ &\text{Coarse:} && \pm\, 2{\cdot}0\,(0{\cdot}004D + k)\end{aligned}$$

where D = dimension in inches involved, and
k = additional allowance depending on location of dimension in the moulding.

Details of the k allowances are given in Table 80.

This table has been altered slightly from the original publication in that the tolerances have been worked out for a range of dimensions D, enabling the limits to be listed instead of calculated for each particular value of D. Fine limits are listed; medium and coarse limits are respectively twice and four times those listed.

In the opinion of the author, this system may be satisfactory for small mouldings, but since the first power of D is used, the tolerances on large parts are excessive. It would be better to use a basic formula $(0{\cdot}004\sqrt{D} + k)$ or $(0{\cdot}004\sqrt[3]{D} + k)$. The approximately equivalent I.S.A. tolerance qualities for a 1 in. dia. are given for interest.

A very comprehensive and well thought out system of tolerances for no less than twenty different types of plastic material (ten types of phenolic, two urea-formaldehyde, three melamine-formaldehyde, one each cellulose acetate, ethyl cellulose, methyl methacrylate and polystyrene) is contained in a brochure published by the Society of the Plastics Industry Inc., New York, under the title *Standards for Tolerance on Moulded Plastic Parts*. This booklet contains a series of curves and charts which are too complicated to reproduce but are in fact simpler to use than the B.P.F. system. Typical examples of the standard S.P.I. Tolerances are as follows (closer and wider tolerances are listed)—

Material	Moulding	Dia. Tolerance in One Tool Part			Cavity Depth Tolerance	
		1 in. Dia.	3 in. Dia.	5 in. Dia.	1 in. Depth	3 in. Depth
Gen. purpose phenolic	Comp.	± 0·0045	± 0·007	± 0·010	± 0·008	± 0·012
High impact phenolic	Comp.	± 0·0055	± 0·008	± 0·011	± 0·010	± 0·014
Alpha-cellulose filled urea	Comp.	± 0·007	± 0·011	± 0·015	± 0·010	—
Polystyrene	Inject.	± 0·004	± 0·005	± 0·0065	± 0·004	± 0·007
Cellulose acetate	Inject.	± 0·004	± 0·0055	± 0·007	± 0·005	± 0·009

Tolerances on Moulded Threads

The same errors as with plain moulding occur with thread moulding, but are usually more serious. To compensate for errors of thread form due to variations in powder shrinkage, resulting in serious pitch errors as high as 0·003 in. per inch, the effective diameter of a screw has to be reduced appreciably (or the nut increased). Allowance can be made for the mean shrinkage by using thread

TABLE 81

B.P.F. Moulding Tolerances: Fine Limits—Values of k

Fine Limits = $\pm \frac{1}{2}(4D + k)$ (Unit = 0·001 in.)			k	Dimension D, in. −0·5	0·51–1·0	1·01–2·0	2·01–3·0	3·01–4·0	4·01–5·0	5·01–6	Equivalent I.S.A. Tolerance Quality D = 1 in.
Dimensions involving one mould part only	Diameters or equivalent		3	2·5	3·5	5	7	9	11	13	10
	Centres	Moulded holes	6	4	5	7	9	11	13	15	11
		Moulded, shrunk, or glued inserts	8	5	6	8	10	12	14	16	11
	Miscellaneous dimensions		7	4·5	5	7	9	11	13	15	11
Dimensions normal to the parting plane, affected by flash thickness (up to 20 sq. in. parting plane area)	Fine powder	Single cavity mould	8	5	6	8	10	12	14	16	11
		Multiple cavity mould	12	7	8	10	12	14	16	18	12
	Medium powder	Single cavity mould	12	7	8	10	12	14	16	18	12
		Multiple cavity mould	16	9	10	12	14	16	18	20	12
	Coarse powder	Single cavity mould	25	13	14	16	18	20	22	24	13
		Multiple cavity mould	35	18	19	21	23	25	27	29	14
	For each additional 10 sq. in. ADD		3	—	—	—	—	—	—	—	—
Dimensions involving two different parts of the mould	Concentricity or misalignment*	Holes	3	2·5	3·5	5	7	9	11	13	10
		Inserts	7	4·5	5	7	9	11	13	15	11
	Wall thickness, in direction normal to the pressing axis		10	6	7	9	11	13	15	17	12
	Other dimensions involving loose pieces of the tool		10	6	7	9	11	13	15	17	12
Bow or warping of plane surfaces			5	3·5	4·5	6	8	10	12	14	11

* Calculate limits on an insert and its moulded hole separately and add tolerances to give full relative tolerance.

moulds with elongated pitch, but it is the variation in shrinkage which causes the main difficulty.

With the state of the art as it is at present, the designer must not expect well-finished, accurate, moulded threads, and as far as possible should help the toolmaker and moulder by calling for moulding only on unstressed threads of short engagement, coarse pitch, and which may be appreciably slack.

2. Sintering

The following limits are recommended in *Mechanical Engineering*, November, 1946*—

	Diameter, in.: Tolerance unit = 0·001 in.					
Dimensions, in.	–1·0	1·01–1·5	1·51–2·0	2·01–2·5	2·51–3·0	3·01–4·0
Length† . . .	± 5	± 10	± 15	± 20	± 25	—
Diameters not sized or coined . . .	± 1·5	± 2	± 3	± 4	—	—
Flange diameters .	± 4	± 6	± 8	± 10	± 14	± 16
Concentricity, dial reading . . .	3	4	5	6	—	—

† Normal to die split and affected by filling of die, compression of powder, etc.

	Flange Thickness, in.		
	–0·25	0·251–0·375	0·376–0·50
Tolerance on flange thickness .	± 0·004	± 0·006	± 0·008

* "Some Consideration in Designing Parts for Powder Metallurgy," by I. J. Donahue, *Mechanical Engineering* (American Society of Mechanical Engineers), Vol. 68, No. 11, November, 1946, page 949.

Tables 82 and 83 list tolerances on self-lubricating bearings.

TABLE 82

LIMITS ON OILITE BUSHES

	Tolerance unit = 0·001 in.		
Dimension	–1·125	1·126–2·125	2·126–
Outside diameter	+ 0·5 + 0	+ 1·0 + 0	+ 2·0 + 0
Inside diameter	– 0 – 0·5	– 0 – 1·0	– 0 – 2·0
Length	± 5·0	± 5·0	± 10·0
Concentricity	2·0–4·0 dial reading		

TABLE 83

LIMITS ON COMPO BEARINGS

Dimension	Tolerance unit = 0·001 in.	
	Up to 2·125 in. dia.	Above 2·125 in. dia.
Plain outside diameter	+ 1·0, + 0	+ 2·0, + 0
Inside diameter	– 0, – 1·0	– 0, – 2·0
Length	± 5	± 7·5
	Up to 1·5 in.	1·51–3·0 in.
Concentricity on dial	3	4
Flange diameter	± 2·5	± 5
Flange thickness	± 2·5	± 5
Spherical outside diameter	± 2·5	± 5
Thrust washer	Up to 2 in.	Above 2 in.
Outside diameter	± 2·5	± 5
Inside diameter	± 2·5	± 2·5
Thickness	± 2·5	± 5
Parallelism	2·0 min.	2·0 min.

RUBBER

1. Rubber Moulding

Sources of Error

Some of the sources of error on normal rubber moulding process are—

Thermal shrinkage.
Volume change during vulcanization.
Mould errors or distortion.
Lack of homogeneity of rubber after calendering or mixing.

Comments on these Errors

Thermal shrinkage analogous to that with castings or with plastic moulding is due to the rubber being moulded at the vulcanizing temperature of about 140° C. The coefficient of expansion of rubber is much higher than that of the steel used for the mould (about 20 times). This source of error can usually be calculated and allowed for.

Chemical changes during vulcanization may give rise to an appreciable shrinkage, which can be allowed for by trial and error.

Mould errors of the usual dimensional type are naturally present, as well as special errors due to distortion of long or large moulds under moulding pressure.

Mixing faults, although unlikely with modern rubber compounding technique, will cause variations of chemical change and to a small degree thermal shrinkage.

In the case of extrusion, considerable variations occur in section dimensions due to lack of uniformity of extrusion pressure, variations in the mix, etc., and a much lower standard of dimensional accuracy must be accepted as compared with moulding.

Representative Limits

On moulded sheets or discs not exceeding 12 in. diameter—

Thicknesses up to 0·2 in.
Precision work: Tolerance = 0·008 in.
Normal work: Tolerance = 0·012 in.

On general sheet or strip—

Normal Tolerance, in.		
Sheet Width	Thickness, in.	
	–0·125	0·126–0·25
0–3 ft.	0·015	0·020
3–6 ft.	0·020	0·025

General Tolerances on Moulded Products. The most accurate work will not give a dimensional accuracy better than ± 5 per cent (1 ± 0·05 in.) without excessive scrap, except as indicated above. For average commercial work ± 10 per cent is to be expected. These percentages refer to a controlling dimension such as thickness of a regular section or the thickness and overall length of an I- or dumb-bell-shaped moulding. It is not possible to quote more precise information applicable to all dimensions of a complicated part, as the shrinkage will be variable in different parts of the moulding.

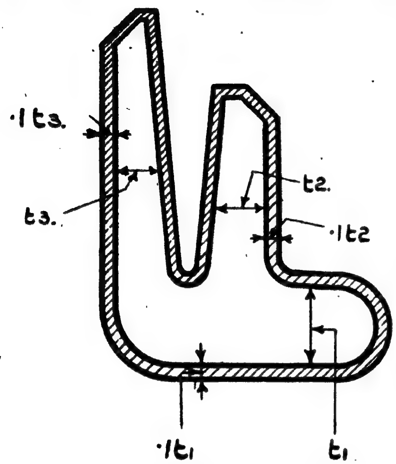


Fig. 83. Tolerance Zones for Typical Rubber Extrusion

General Tolerances on Extruded Rubber Sections. On fairly regular sections up to 2 in. maximum dimension, an approximate idea of the tolerance zone for the section can be gained by taking the tolerance on the various average thicknesses of different parts of the section as ± 10 per cent, and blending in these zones to form the remainder of the complete zone for the section. (See Fig. 83.)

2. Rubber Cutting

Rubber can be machined by those skilled in the art to very close limits. Rings can be cut on lathes with a section tolerance of 0·002 in. for a section thickness of up to 0·25 in. It is quite usual for rubber moulders to machine slabs of rubber into the shape of a moulding for experimental work, and accuracy equivalent to that of the moulding process itself can be achieved. Certain types of rubber product are also ground to size; for example, typewriter rollers, or the ends of flexible hose assemblies before fitment of the end ferrules or connections can be held to 0·005 in. by grinding.

MISCELLANEOUS

1. Flame Cutting

Sources of Error

Sources of error with flame cutting are due to—

Cam or pantograph errors.

Flame adjustment (width, length, temperature, etc.).
Thermal dilatation of the metal.
Melting of metal.
The rate of cutting or flame movement.

Comments on these Errors

The copying or guiding mechanism tracing the flame movement is subject to errors of wear, adjustment, etc., and obviously the accuracy of reproduction can be no better than the master and the pantograph or equivalent mechanism will allow.

The flame itself will vary in magnitude, temperature, etc., due to variations in gas supply pressure, etc., to a much greater degree than any normal tool. As the metal heats up during cutting, severe irregular expansion may take place, requiring compensation on the pattern.

The actual melting of the metal will vary due to heat conduction, variations in the local mass of metal, and to the rate of heat input, (i.e. rate of flame movement).

The fact that a fair degree of accuracy can be achieved is due to close control of all variables, particularly the flame itself.

Representative Limits

Since the surface as cut is relatively rough, the tolerance on the mean cut is much less than that if the extreme variation in size is taken into account (i.e. the surface roughness). For many processes the roughness of the surface is less important than the position of the mean, and can be considered separately. The mean cut can conveniently be taken as that determined by laying a straight edge along the cut surface, although this is not the mean in a geometrical sense. The surface first exposed to the flame is usually melted excessively locally and may exceed the average error of the remainder of the cut.

Tolerance on Mean Cut. This can be expressed as the maximum deviation of the mean profile from the nominal dimension (i.e. assuming the surface roughness is ignored), as measured with a straight edge.

Thickness of Metal Cut, in.	Max. Deviation on any One Cut, in.
–0·5	± 0·025
0·51–1·5	± 0·05
1·51–3	± 0·08
3·01–6	± 0·1
6·01–	± 0·12

These tolerances include lack of squareness in the cut surface.

On a strip or sheet produced in two cuts, the limits are thus twice those given above.

TOLERANCE ON SURFACE FINISH. The error on the surface itself is over and above these tolerances, and applies to the mean size, being below the surface determined by a straight edge along the cut.

Thickness of Metal Cut, in.	Max. Deviation of Surface of Cut below Straight Edge, in.
-1·5	− 0·06
1·51–6	− 0·1
6·01–	− 0·15

2. Continuous Centreless Ground Steel Rod

This is commercially available up to about 1 in. in diameter in various grades of accuracy as follows—

(i) $\pm$ 0·002 in.
(ii) $\pm$ 0·001 in.
(iii) $\pm$ 0·0005 in.

3. Pressing

It is to be regretted that information on the subject of the accuracy which is obtainable by the pressing processes, both cold and hot, is simply not available.* Undoubtedly the accuracy which is in fact achieved is very closely related to the tools which produce the parts; a great deal of study and experience is put into press tools, and by the use of several sets of tools very satisfactory results can be obtained.

Thus to table any tolerances which can be worked to could only be done in relation to the particular type of tool and press being used, and these are legion.

Familiar examples of press work of high accuracy are the ordinary small paint tin, with a press on top with fully interchangeable interference limits; a motor car wheel hub cap, closely fitted to a retaining device on the wheel; transformer laminations; radio condenser pressings accurately blanked to about 0·003 in.; radio valve parts, etc.

Hot pressings, such as wheels, chassis frames, etc., obviously cannot be produced with great accuracy, since thermal distortion

* A little information is contained in *Production Processes—their Influence on Design* (see Bibliography).

and contraction will affect the finished dimensions, and usually such processes are applied to thick sheets, again affecting blanking accuracy.

Since the pressing process involves overstrain of the metal, the ductility of the metal will affect the final result, other factors being equal. Thus it is easier to achieve accuracy with brass or soft aluminium than sheet steel or high-tensile light alloy.

4. Spinning

The spinning process, like that of pressing, introduces too many variable factors for the tolerance applicable to the process to be specified simply. Diameters can be spun to an accuracy from $\pm$ 0·015 in. on small work to $\pm$ 0·06 in. on large parts. More precise information depends on the shape of the component.

Further information on the process is contained in *Production Processes—their Influence on Design* (see Bibliography).

CHAPTER VII

SCREW THREAD TOLERANCES

1. Introduction.

THE average engineer is usually not concerned with the determination of thread tolerances themselves, but only with the application of given fits from published specifications of thread tolerances, prepared by experts and issued by Government bureaux, etc. However, a knowledge of problems involved in determining thread limits is necessary for the proper selection of thread fits, and it is safe to say that a better knowledge would, in many cases, enable thread tolerances to be widened without affecting performance in any way.

Thread production technique has improved enormously in the last twenty-five years owing to the popularity of thread rolling, grinding, and milling. Pitch errors due to machine lead screw faults have been reduced as the lead screws themselves improve. Thread form, with the wide use of projection inspection, is of a high standard.

The fact remains, however, that comparatively wide deviation of form and fit do not noticeably affect thread strength, except perhaps under fatigue conditions, and in many engineering specifications the thread quality requirements are too severe; the object of this section is to acquaint the engineer with the elements of thread tolerances in the hope that he may be helped to apply them wisely.

2. Thread Forms and Tolerance Zones

We are not concerned with the exact details of the thread form, its dimensions, angles, etc., but with the errors and tolerances involved. The three main types of thread (B.S. Whitworth, Metric, and American National) are illustrated in Figs. 84, 85, and 86, with their tolerance zones.

The *Whitworth* form (Fig. 84) is similar as regards tolerance zones to the B.S. Cycle and B.A. Threads, both with rounded roots and crests but differing thread angles. It will be noted that the basic or theoretical thread form is the screw maximum and nut minimum. The lower limit of screw profile is defined by the screw tolerance zone. The upper limit of nut profile is defined by the nut tolerance zone except that the maximum major nut diameter is not specified. This is because the exact maximum size of this diameter is not important from a strength point of view, since it cannot

exceed what will be produced by the tap, should the hole be tapped, and if screw cut, some form of undercut of any important depth could not conceivably be produced.

The German Whitworth thread which is widely used in that country is similar to the British Whitworth, except that there is a minimum clearance between the crest of bolt and the basic thread

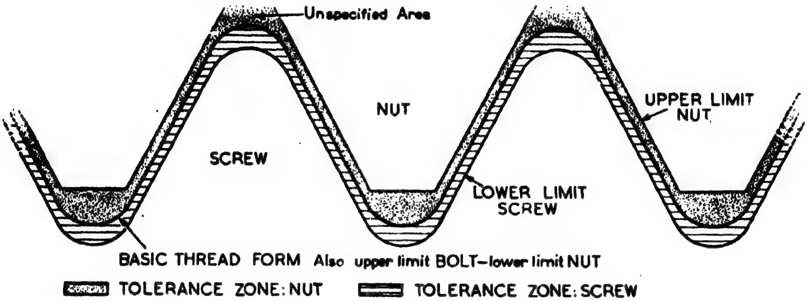


FIG. 84. THREAD TOLERANCE ZONES—B.S. WHITWORTH, B.A. AND B.S. CYCLE

form, i.e. the bolt diameter is always slightly less than nominal size. Also the tolerance on the major diameter of the bolt is sufficiently generous to cause truncation of the crest radius.

The *Metric* (Système International) form (Fig. 85) differs from the Whitworth, apart from the angle, in having a flat crest and an unspecified root, which can be either flat or radiused. The radius if

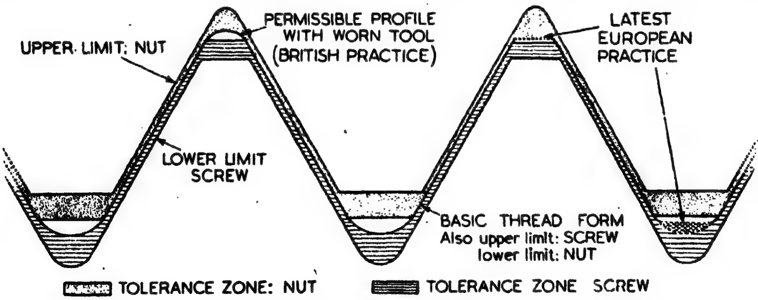


FIG. 85. THREAD TOLERANCE ZONES—B.S. METRIC THREAD

used is too small to have any beneficial effect on thread fatigue strength. Again the basic thread flank profiles of screw and nut coincide at maximum screw and minimum nut. According to the B.S. metric form (B.S. 1095), the maximum minor diameter of screw and minimum major diameter of nut are supposed to occur when the two basic size flanks are blended in with radii as shown.

In actual practice the tool which originally has a small radius will not wear into a regular and larger radius, but will become more and more flat, filling up the crest clearance. Latest European practice acknowledges this, and the profile with a worn tool can approach the basic thread form (i.e. with or without any crest clearance). The minimum minor diameter of the screw and maximum major

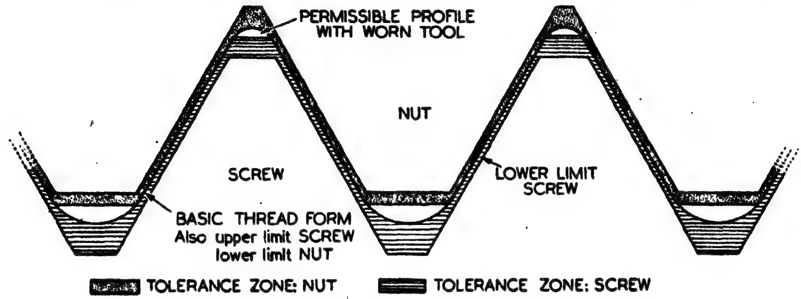


Fig. 86. Thread Tolerance Zones—American National Thread

diameter of the nut are specified in terms of the minimum radius in the B.S. Metric form; and as a flat as regards the screw, and unspecified for the nut, in the I.S.A. standard proper.

The *American National* thread (Fig. 86) is generally similar to the metric form, the screw and nut being "basic." The tolerance zone for the screw determines the minimum condition. The minor screw diameter lies between the radius joining the two basic flanks and a flat of $\frac{1}{8}p$. As pointed out above, there seems no reason why

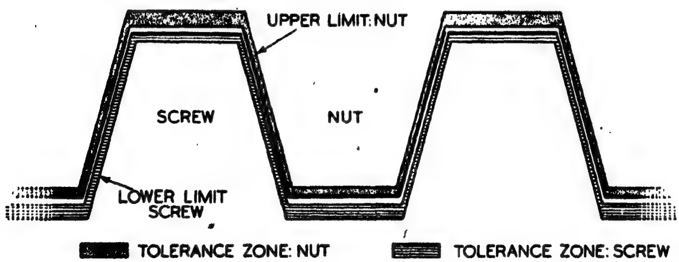


Fig. 87. Thread Tolerance Zones—29° Acme Thread

the tool should not wear until the "radius" becomes a flat of $\frac{1}{4}p$, coinciding with the nut minimum. Similar remarks apply to the nut major diameter radius. The maximum major diameter of the nut is the apex of the theoretical triangle.

Other common forms of thread are the *Acme* (Fig. 87) and *Buttress* (Fig. 88). Apart from the special profiles involved these

have tolerance zones similar to more standard threads. The Acme thread usually fits closely on the flanks and has appreciable clearance at crest and root. There is, however, a new range of American "centralizing" Acme threads which fit fairly closely on the major diameter as well as on the effective diameter. Buttress threads are

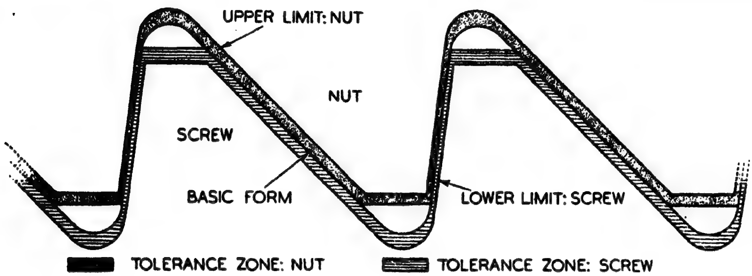


FIG. 88. THREAD TOLERANCE ZONES—BUTTRESS THREAD

intended to fit on one flank only, the other flank and usually the major diameter being clear, and the minor diameter well clear.

3. Types of Fit

A *clearance fit* is used on slack threads of the type used in commercial black nuts and bolts, carriage bolts, etc. Here large tolerances are necessary and the minimum clearance takes care of imperfections of manufacture, ensuring that the threads always assemble. Clearance fits are listed for A.N. (Class 1) threads (see Fig. 89), but not B.S. threads.

TABLE 84

Thread	Effective Diameter Tolerance (unit = 0·001 in.)		
	Fine Grade	Medium Grade	Coarse Grade
¼ in. × 20 t.p.i. Whitworth	2·6	3·9	5·8
¼ in. × 20 t.p.i. A.N.	2·6 (Cl. 3)	3·9 (Cl. 2)	*
6 mm × 1 mm D.I.N.	2·7	4·0	6·6
1 in. × 8 t.p.i. Whitworth	4·5	6·8	10·2
1 in. × 8 t.p.i. A.N.	5·4 (Cl. 3)	7·4 (Cl. 2)	*
24 mm × 3 mm D.I.N	4·6	6·9	11·4

* Class 1 is used here, but it has large minimum clearance.

Normal fits have the minimum nut and maximum bolt effective diameters both zero, with various grades of tolerance on each. B.S. threads are listed as close, medium, and free, with a tolerance

quality progression of 1, 1½, 2½. A.N. threads have two normal fits, Class 3 and Class 2, with a tolerance progression of 1, 1½ (Fig. 89). The German metric threads (D.I.N. 3012) have three normal fits, fine, medium, and coarse, with a tolerance progression of 1, 1½, 2½. The intermediate fit is the one intended for general use in both

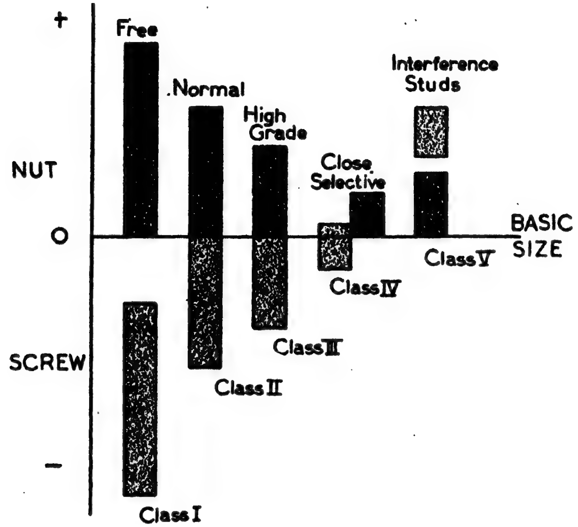


Fig. 89. American National Thread Tolerances (Effective Diameter)

cases. Table 83 shows the relative magnitude of various thread system tolerances.

A *transition* fit is listed in the A.N. Class 4 thread system only. The tolerance on this fit is the progression Class 2, Class 3, Class 4: 1½, 1, ½. The maximum interference is small (0·0007 in. for an 8 t.p.i. thread).

Interference fits are listed in the Class 5 A.N. threads, and in other special threads for studs (e.g. B.S. 1171).

4. Selection of Fit

General

In selecting the correct fit and tolerance grade of a screw thread for normal duty, the following factors must be considered—

The minimum depth of engagement for strength.
The importance of clearance.

The minimum depth of engagement on extreme limits is generally not allowed to depart from the maximum or theoretical amount by

more than 20–30 per cent. It has been shown in recent years that a comparatively large reduction in engagement has little effect on thread strength, at least for ordinary bolts and nuts, and a reduction of engagement to 50 per cent is often acceptable. In the case of large diameter fine pitch threads the limiting factor on thread strength is rarely shearing, but jumping of the threads due to the expansion of the nut, and a reduction in engagement will make this occur more easily. Engagement is reduced as the tolerances become coarser, but medium tolerances will give adequate strength in all cases and close tolerances are rarely justified on these grounds. For all normal nut and bolt problems, even on precision equipment, medium bolts with free or coarse nuts will give adequate performance in the smaller sizes up to $\frac{3}{4}$ in. Truncation of the thread, either deliberately by tapping the major diameter of the screw or indirectly by using an enlarged tapping drill, will obviously reduce the thread engagement and for adequate performance further liberties with the tolerances should not be taken—i.e. use medium nuts and bolts for normal good quality work.

Apart from the desirability of having adequate clearance on black nuts and bolts, already mentioned, there are many cases when the relatively large clearances which will result on the average from using medium or coarse fits are an advantage. For example, where the concentricity of a thread with a plain diameter is important, adequate clearance will cover small errors of concentricity and facilitate assembly. The probability that the two threads were on extreme limits (i.e. size and size) and eccentric at the same time is very small. Where rapid assembly of two threads is essential, particularly in the field (e.g. a theodolite tripod screw, a camera lens screw), slack threads are an advantage, and usually strength is of secondary importance.

Close-fitting threads, although used automatically by some designers on high-class products because they appear to be the correct thing, are rarely necessary on any design except where play is to be eliminated or minimized. An obvious example is in "motion" screws, as on a lathe, or when a screw and nut must carry alternating loads without being locked up tight. When fine clearances are required the screw accuracy is important for other reasons (e.g. pitch accuracy) and the method of manufacture automatically produces a thread to close limits. But to specify close tolerances on a big end bolt of a connecting rod because the bolt is a highly stressed one is wrong.

The American Class 4 transition fit is a refinement on the usual close fit in that by selection the finest fit accuracy can be achieved, and this has some utility in the few cases where such a fine running

fit is required (a camera focusing screw is an example where backlash must be minimized or even eliminated).

It is perhaps better practice to make the nut adjustable, as, for example, by splitting it and rotating the two parts relatively to eliminate backlash.

Interference fits are invariably on studs, and this will be dealt with below.

Studs

It is a curious thing that although studs have been used throughout the world for years, no well-established practice has been standardized to ensure that the inner end of the stud is driven home and stays there. There are three ways of keeping the stud tight (apart from fixing it)—

Allow it to foul on its lower end on the bottom of the hole.

Allow the threads to foul on the run-out at the end of the stud threads or bottom of the tapped hole.

Use interference fit threads.

It is not sufficient to allow the stud to bottom as it will inevitably loosen in service and unscrew when the nut is removed.

The practice of allowing the threads to foul, usually at the top or outer end of the tapped hole, is very widespread and works reasonably well. Studs do work loose, particularly in light alloy casings, and the degree of clamping on the stud is not considered adequate for high-class production. In addition, the interference of the thread throws a burr up round the edge of the tapped hole which is often inconvenient.

The best practice is to use interference fit threads. This can be done by using oversize studs or undersize holes. All the usual difficulties of interference fits are introduced, and to avoid selective assembly very close tolerances on the effective diameters are necessary. The A.N. threads list Class 5 tolerances giving a true interference fit under all conditions but the tolerances are even finer than the "close" type of fit. The stud tolerance is approximately the same as for Class 4—i.e. about half Class 3 or B.S. "Close"; the hole tolerance is about 50 per cent greater than the stud tolerance. It is obvious that such limits can only be achieved with the best quality of thread rolling or by grinding, and by tapping with ground thread taps.

There is a British Standard (B.S. 1171) for high duty studs for aero engines listing a few sizes up to $\frac{5}{8}$ in. dia., 16 t.p.i. Effective diameter tolerances for both stud and hole are B.S. "Close." To enable standard size threads to be used on the stud, the interference

is obtained by using undersized holes tapped with special taps; in fact, three sizes of tap for each thread are listed. Fig. 90, reproduced from B.S. 1171, shows the effective diameter tolerance zones for studs, holes, and taps.

The American Standard uses standard holes with an oversize stud, the two ends of the stud being to a different tolerance but readily distinguished notwithstanding. The inconvenience of the British system must be considerable, especially as "oversize-undersize" taps are necessary for repair work, but is considered less than in having the stud ends different. In aero engine practice, where the studs are thread rolled, this point is important, since a change in machine setting is necessary to deal with the different effective diameter sizes.

For normal commercial work, a satisfactory compromise is to use a higher transition class of fit with reasonable tolerances ("close" stud; "medium" hole), and use selective assembly to reject the slackest fits. If replacement studs are supplied by grading to the large size, the necessary allowance for enlargement of the hole in service can be made.

A reasonable basis for this in the case of B.S. threads is to use tapped holes to standard medium tolerances, and to make the stud to "close" limits but increase the minimum effective diameter by the magnitude of the "close" tolerance itself. Fig. 91 makes this more clear, and it will be seen that since the close tolerance is two-thirds of the medium tolerance, the maximum and minimum fits are as follows—

Maximum interference = 2 × close tolerance (0·0066 in. on a $\frac{1}{2}$ in. B.S.F. stud).

Maximum clearance = $\frac{1}{2}$ × close tolerance (0·0017 in. on a $\frac{1}{2}$ in. B.S.F. stud).

Mean fit = an interference of $\frac{3}{4}$ × close tolerance (0·0025 in. on a $\frac{1}{2}$ in. B.S.F. stud).

If this procedure is adopted, the major diameter of the stud should stay at the normal "close" tolerance or it can without harm be amended to the "medium" equivalent; the minor diameter should be increased along with the effective diameter. Under these circumstances, the thread form will not be perfect and the full crest radius will not be achieved.

In order to prevent fouling at the thread root, the tapped hole should be drilled with an oversize tapping drill; the minor diameter should then be well clear of the stud minor diameter and this will allow the metal displaced by the interference fit to flow properly. A convenient basis for the hole minor diameter limits is to enlarge

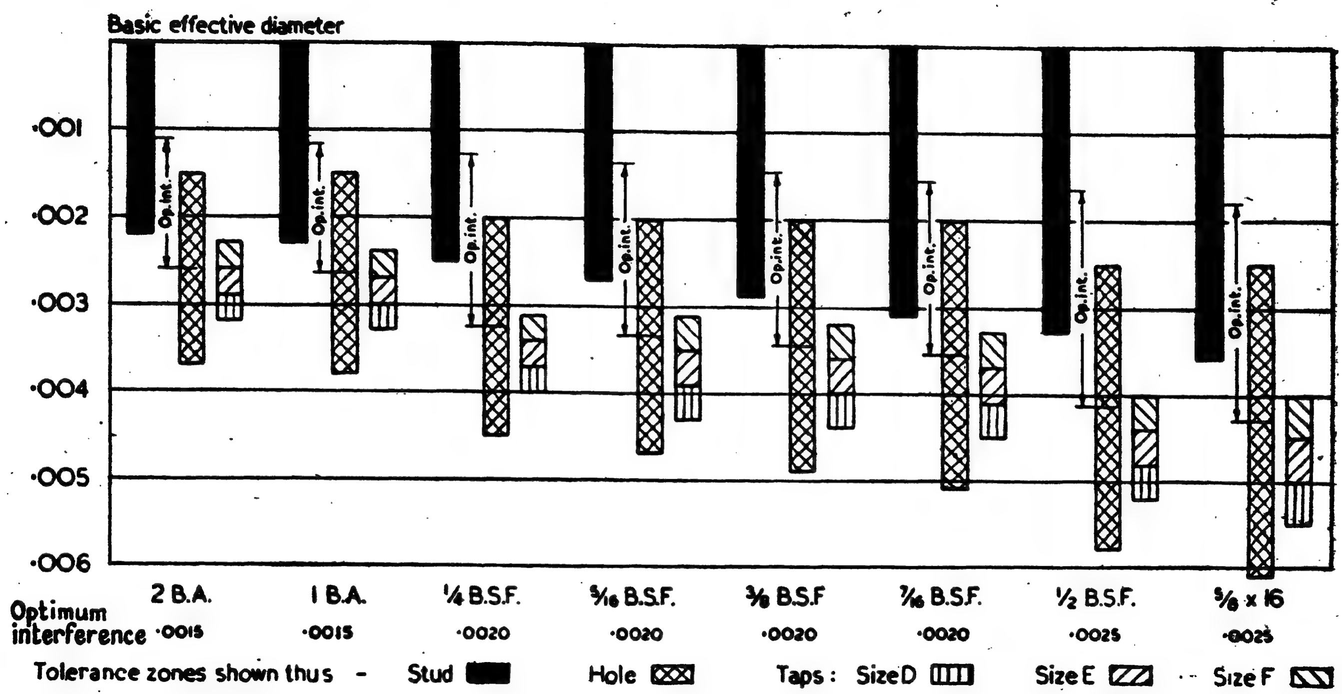


FIG. 90. B.S. 1171. EFFECTIVE DIAMETER TOLERANCE ZONES FOR STUDS, HOLES, AND TAPS

the hole by the amount of the standard tolerance on the minor diameter itself. For example, the limits on the minor diameter of a 0·75 in. × 10 t.p.i. B.S.W. thread, medium fit, are 0·6940/0·6220 (tolerance 0·027 in.); with a stud on minimum major diameter the depth of engagement is about 75 per cent of the full amount. If the limits on the minor nut diameter are 0·6490 + 0·027 = 0·6760/0·6220 + 0·027 = 0·6490, the clearance will be adequate and the depth of

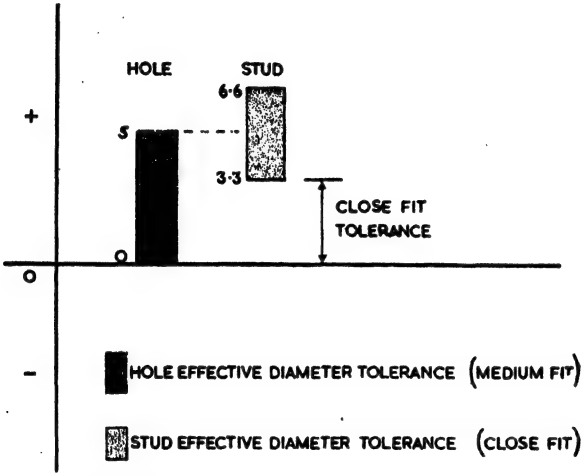


FIG. 91. STUD LIMITS (TO SCALE FOR ½ IN. B.S.F.)

engagement not less than 50 per cent, which is satisfactory for an interference fit of this nature.

If due allowance is made for different lengths of thread, for different metals (good practice: 1·0 diameter in steel, 1·5 diameters in cast iron, 2·0 diameters in light alloy, even 3 diameters in important studs in magnesium alloy), these limits should be applicable to all cases.

Truncation

Truncation of the male thread by reducing the major diameter (see B.S. 84), or the female threads by using an enlarged tapping drill (see B.S. 1157), as mentioned above under "Studs," is common practice in some factories. This has very little effect on thread strength, in spite of reduced engagement, but has no fundamental effect on the thread tolerances. Advantages can be taken, however, on certain classes of work to widen the major bolt or minor nut limits without altering the effective diameter limits. An example

of this is in the production of threads on drawn tubing where the major diameter may be considerably undersize; standard limits would not be possible.

5. Manufacturing Errors in Screw Production

Apart from errors which arise in normal machining or drilling and which occur in screw thread production as well, there are certain special types of error more or less particular to threads.

These are errors of form and errors of pitch.

The first errors of form are due mainly to imperfections or wear in the threading equipment (chaser, tap, etc.), but in the case of screw cutting with a single point tool obviously the thread form, particularly with Whitworth threads, will rarely be correct. These errors are not in themselves particularly important, although angle errors cause the threads to contact at points instead of along the flanks; this is equivalent to an increase in effective diameter of the screw or a reduction in the thread clearance. On some highly stressed threads the root radius on male threads is important, and errors in form at this point adversely affect thread fatigue strength; in the case of threads with flat or sharp roots, errors may actually increase the thread fatigue strength by the sharp edge on the tool wearing into a smooth radius.

Errors of overall size are due to the chasers on a die box or the tap being worn or set wrong, so that the thread is correct in all respects except that the nominal diameter is oversize or undersize.

Finally, errors of pitch, usually associated with form errors as well, are due to faults in the machine lead screw or the screw of the machine grinding the die or tap. With unground taps and dies, distortion in hardening will cause appreciable errors. Unless the screw is a "motion" screw used for accurate translations, as in a micrometer, pitch errors are only important over the length of thread actually used on the nut; gauging practice takes this into account, as will be seen later. Errors in pitch, like errors of form, are equivalent to an increase in effective diameter tolerance (i.e. increase in *size* of screw and decrease in *size* of nut). The relationship is determinable by simple trigonometry, and all modern thread systems publish the formulae and tables derived from them for use by inspectors and screw thread equipment makers.

6. Derivation of Limits and Tolerances

In the following section the derivation of limits and tolerances on common thread systems is explained, using the following terminology—

D = major diameter of thread,
L = length of engagement (nut length, or 10 pitches for special threads unless otherwise stated),
p = pitch (length),
h = basic height of thread.

Due regard should be paid to the use of correct units.

(a) *B.S. Whitworth Form Threads* (B.S. 84)

TOLERANCE ON EFFECTIVE DIAMETER

$$\text{Medium tolerance} = 0{\cdot}002\sqrt[3]{D} + 0{\cdot}003\sqrt{L} + 0{\cdot}005\sqrt{p}$$
$$\text{Close tolerance} = \tfrac{2}{3} \times \text{medium tolerance}$$
$$\text{Free tolerance} = 1{\cdot}5 \times \text{medium tolerance}$$

TOLERANCE ON MAJOR DIAMETER OF BOLT

$$\text{Tolerance} = \text{effective dia. tolerance} + 0{\cdot}01\sqrt{p}$$

In the special case of truncated threads the limits and tolerance are as follows—

$$\text{The degree of truncation} = 0{\cdot}147835p$$
$$\text{Tolerance} = 0{\cdot}052p + 0{\cdot}003 \text{ in.}$$

TOLERANCE ON MINOR DIAMETER OF BOLT

$$\text{Close tolerance} = \text{close effective dia. tolerance} + 0{\cdot}013\sqrt{p}$$
$$\text{Medium tolerance} = \text{medium ,, ,, ,,} + 0{\cdot}02\sqrt{p}$$
$$\text{Free tolerance} = \text{free ,, ,, ,,} + 0{\cdot}02\sqrt{p}$$

TOLERANCE ON MINOR DIAMETER OF NUT

$$\text{Tolerance (all fits)} = 0{\cdot}2p + 0{\cdot}004\ (p = \rightarrow 26 \text{ t.p.i.})$$
$$= 0{\cdot}2p + 0{\cdot}005\ (p = 24 - 22 \text{ t.p.i.})$$
$$= 0{\cdot}2p + 0{\cdot}007\ (p = 20 \text{ t.p.i.} \rightarrow)$$

For truncated nuts, the upper limit is as given above, the lower limit being equivalent to a truncation of $0{\cdot}147835p$.

$$\text{Tolerance (all fits)} = 0{\cdot}052p + 0{\cdot}004\ (p = \rightarrow 26 \text{ t.p.i.})$$
$$= 0{\cdot}052p + 0{\cdot}005\ (p = 24 - 22 \text{ t.p.i.})$$
$$= 0{\cdot}052p + 0{\cdot}007\ (p = 20 \text{ t.p.i.} \rightarrow)$$

(b) *B.A. Threads* (B.S. 93) (dimensions in mm)

TOLERANCE ON EFFECTIVE DIAMETER.

$$\text{Tolerance} = 0{\cdot}08p + 0{\cdot}02$$

TOLERANCE ON MAJOR DIAMETER OF BOLT.

Tolerance $= 0{\cdot}15p$

TOLERANCE ON MINOR DIAMETER OF BOLT.

Tolerance $= 2(0{\cdot}08p + 0{\cdot}02)$

TOLERANCE ON MINOR DIAMETER OF NUT.

Tolerance $= 0{\cdot}35p$ (previously $0{\cdot}15p$).

TOLERANCE ON MAJOR DIAMETER OF NUT.

Tolerance $= 2(0{\cdot}08p + 0{\cdot}02)$

(c) *American National Threads* (*H*28—1944)

TOLERANCES ON EFFECTIVE DIAMETER OF STANDARD THREAD SERIES (CLASSES I–IV)

$$\text{Tolerance} = a \times \left(\frac{p}{0{\cdot}05}\right)^n + b$$

where
$$\begin{aligned} a &= 0{\cdot}0045 \text{ (Class I)} \\ &= 0{\cdot}003 \text{ (Class II)} \\ &= 0{\cdot}002 \text{ (Class III)} \\ &= 0{\cdot}001 \text{ (Class IV)} \\ n &= 0{\cdot}6 \text{ when } p \text{ is finer than 20 t.p.i.} \\ &= 0{\cdot}9 \text{ when } p \text{ is coarser than 20 t.p.i.} \end{aligned}$$

(Note: $\left(\frac{p}{0{\cdot}05}\right)^n = 1$ when $p = 20$ t.p.i.)

$b =$ sum of 2 effective diameter tolerances on X class gauges (as published in H28)

TOLERANCES ON EFFECTIVE DIAMETER OF SPECIAL THREADS (CLASSES I–IV)

$$\begin{aligned} \text{Tolerance, Class I} \quad &= 0{\cdot}002\sqrt{D} + 0{\cdot}002L + 0{\cdot}02\sqrt{p} \\ \text{,,} \quad \text{,,} \quad \text{II} \quad &= 0{\cdot}002\sqrt{D} + 0{\cdot}002L + 0{\cdot}01\sqrt{p} \\ \text{,,} \quad \text{,,} \quad \text{III*} &= 0{\cdot}002\sqrt{D} + 0{\cdot}002L + 0{\cdot}005\sqrt{p} \\ \text{,,} \quad \text{,,} \quad \text{IV} \quad &= 0{\cdot}001\sqrt{D} + 0{\cdot}001L + 0{\cdot}0025\sqrt{p} \end{aligned}$$

TOLERANCE ON MAJOR DIAMETER OF BOLT (CLASSES I–V)

Tolerance, Class I $= 2 \times$ Class I effective dia. tolerance

,, ,, II $= 2 \times$ Class II effective dia. tolerance

(except for Class II, threads on hot rolled material, where

Tolerance $= 2 \times$ Class I effective dia. tolerance)

* For the special 8, 12, and 16 t.p.i. and extra fine series, the tolerance for Class III is $0{\cdot}7 \times$ Class II tolerance.

Tolerance, Classes III, IV, and V = 2 × Class II effective dia. tolerance.

Tolerance on Minor Diameter of Nut (Classes I–IV)

This is determined by setting geometrical limits to thread form; which gives the following—

Minimum minor diameter of nut = min. nut effective dia. − $\frac{2}{3}h$.
Tolerance = between $\frac{2}{3}h$ (small sizes) and $\frac{1}{6}h$ (large sizes).

Determination of Deviation or Allowance for Class I and Class IV Fits

This is determined arbitrarily according to Table 85.

TABLE 85

Allowances on A.N. Class I and IV Fits
(Tolerance unit = 0·001 in.)

T.p.i.	Class I Screw, Maximum	Class IV Screw, Maximum
80	− 0·7	
72	− 0·7	
64	− 0·7	
56	− 0·8	
48	− 0·9	
44	− 0·9	
40	− 1·0	
36	− 1·1	
32	− 1·1	
28	− 1·2	+ 0·2
24	− 1·3	+ 0·3
20	− 1·5	+ 0·3
18	− 1·6	+ 0·3
16	− 1·8	+ 0·4
14	− 2·1	+ 0·4
13	− 2·2	+ 0·4
12	− 2·4	+ 0·5
11	− 2·6	+ 0·5
10	− 2·8	+ 0·6
9	− 3·1	+ 0·6
8	− 3·4	+ 0·7
7	− 3·9	+ 0·8
6	− 4·4	+ 0·9
5	− 5·2	+ 1·0
4½	− 5·7	+ 1·1
4	− 6·4	+ 1·3

Class V Fit

The derivation of the tapped hole and stud limits is arbitrary to give the interferences shown in Figs. 92 and 93.

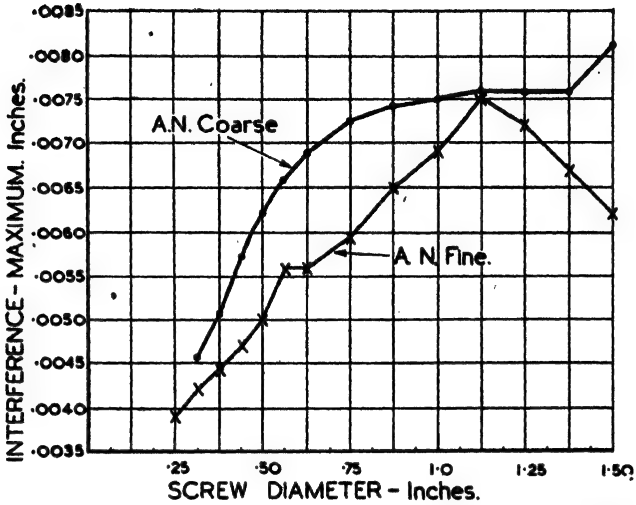


FIG. 92. A.N. CLASS V TOLERANCES—MAXIMUM INTERFERENCE (EFFECTIVE DAMETER)

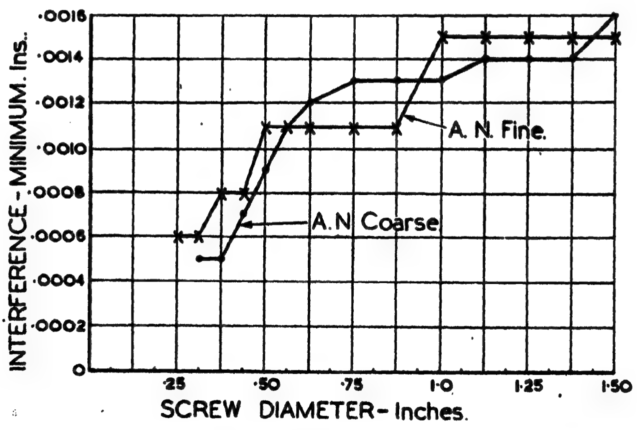


FIG. 93. A.N. CLASS V TOLERANCES—MINIMUM INTERFERENCE (EFFECTIVE DIAMETER)

(*d*) *Metric Threads* (*D.I.N.* 3012)

TOLERANCE ON EFFECTIVE DIAMETER (unit = 0·001 mm)

$$\text{Fine tolerance} = 67\sqrt{p}\begin{cases} +\ \ 0 \text{ (dia. 0–80)} \\ +\ \ 9 \text{ (dia. 84–119)} \\ +\ 19 \text{ (dia. 124–149)} \end{cases}$$

$$\text{Medium tolerance} = 100\sqrt{p}\begin{cases} +\ \ 0 \text{ (dia. 0–80)} \\ +\ 16 \text{ (dia. 84–119)} \\ +\ 32 \text{ (dia. 124–149)} \end{cases}$$

$$\text{Coarse tolerance} = 167\sqrt{p}\begin{cases} +\ \ 0 \text{ (dia. 0–80)} \\ +\ 25 \text{ (dia. 84–119)} \\ +\ 52 \text{ (dia. 124–149)} \end{cases}$$

TOLERANCE ON MAJOR DIAMETER OF BOLT

Fine tolerance = $0{\cdot}05\sqrt[3]{D}$

Medium and coarse tolerances are derived arbitrarily from tolerances on rolled bars, but approximately = $0{\cdot}12\sqrt{D}$.

TOLERANCE ON MINOR DIAMETER OF BOLT

Fine tolerance = 2 × fine effective dia. tolerance
Medium and coarse tolerance = medium effective dia. tolerance + coarse effective dia. tolerance

(This is approximately equal to $267\sqrt{p}$, but the small corrections on larger diameters must be added.)

TOLERANCE ON MINOR DIAMETER OF NUT

Fine tolerance = fine effective dia. tolerance

The medium and coarse tolerances have been determined on an arbitrary basis to control thread engagement, but the tolerances are almost the same as for the corresponding major diameter (i.e. tolerance = $0{\cdot}12\sqrt{D}$ approx.).

(*e*) *B.S. Metric Threads* (*B.S.* 1095)

TOLERANCE ON EFFECTIVE DIAMETER (unit = 1 mm)

Medium tolerance = $0{\cdot}0173\sqrt[3]{D} + 0{\cdot}0136\sqrt{L} + 0{\cdot}0314\sqrt{p}$
Close tolerance = $\frac{2}{3}$ × medium tolerance
Free tolerance = 1·5 × medium tolerance.

(The above formula is identical with that for B.S. Whitworth threads with the constants suitably converted.)

Tolerance on Major Diameter of Bolt

Tolerance $= 0{\cdot}05 + 0{\cdot}108p$

Tolerance on Minor Diameter of Nut

Tolerance $= \frac{1}{4}h = 0{\cdot}1624p$.

7. Screw Thread Gauging Practice

General Considerations

The fundamental principle of the best modern practice in screw thread gauging is—

The "Go" gauge should check as much of the thread at once as practicable.

The "Not-Go" gauges should each check one element of the thread only.

In practice the "Go" gauge is "full form"—i.e. has the theoretical profile in all details, and is of a length at least equal to the length of engagement of the mating thread (standardized at nut width, or about 10 pitches in special threads). Although for a bolt this gauge should strictly be a ring gauge to eliminate oval threads, adjustable caliper gauges are much to be preferred as they can be made with greater accuracy, and used with greater economy since wear can be compensated for. For female threads, screw plug gauges are always used.

The principal "Not-Go" gauge is one for the effective diameter only, the crests and roots being cleared, and should be of the caliper type to measure the least diameter. "Not-Go" ring gauges are often used but are not effective on an oval thread.

The second "Not-Go" gauge for a bolt should be a plain gap gauge and check the minimum major diameter, or degree of truncation, as this affects thread depth of engagement. For a nut, a plain "Not-Go" plug gauge is used to check the minimum minor diameter and degree of truncation for the same reason.

On threads with root and crest clearance (A.N. and Metric), however, if the full form of male or female thread were on the full form gauge the two gauges would not mate, and since slight contact at root or crest might reject a perfectly good thread, it is preferable not to risk this and to clear the gauge at the particular place of clearance. Thus for the bolt, the full form "Go" gauge is cleared at the screw major diameter (crest), and for the nut at the minor diameter (root). The diameter involved is then gauged separately, i.e. for the bolt, a "Go" gap gauge for major diameter; for the nut, a "Go" plug gauge for minor diameter.

Current practice in Britain, U.S.A., and Germany is shown in

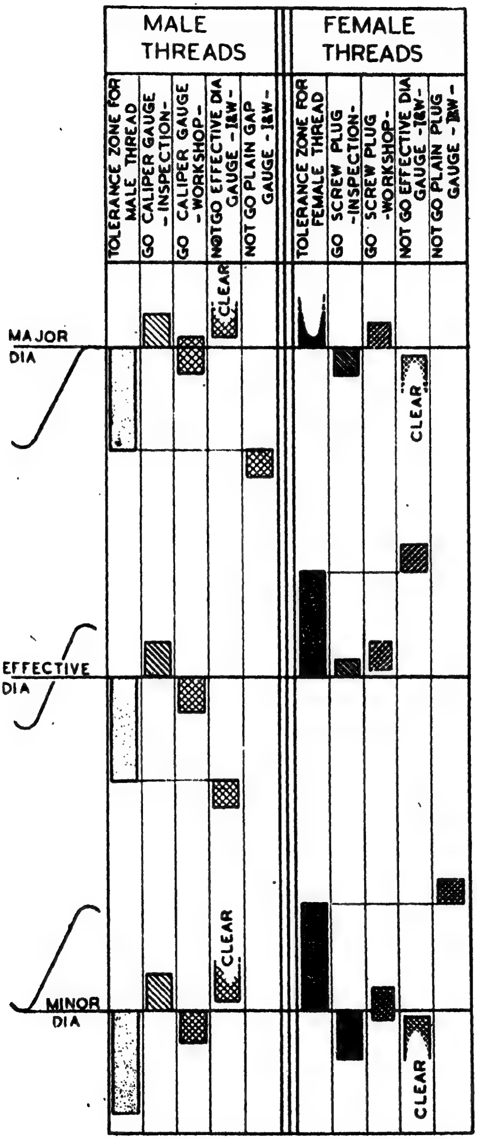


FIG. 94. THREAD TOLERANCE GAUGING DIAGRAM—B.S. WHITWORTH FORM THREADS

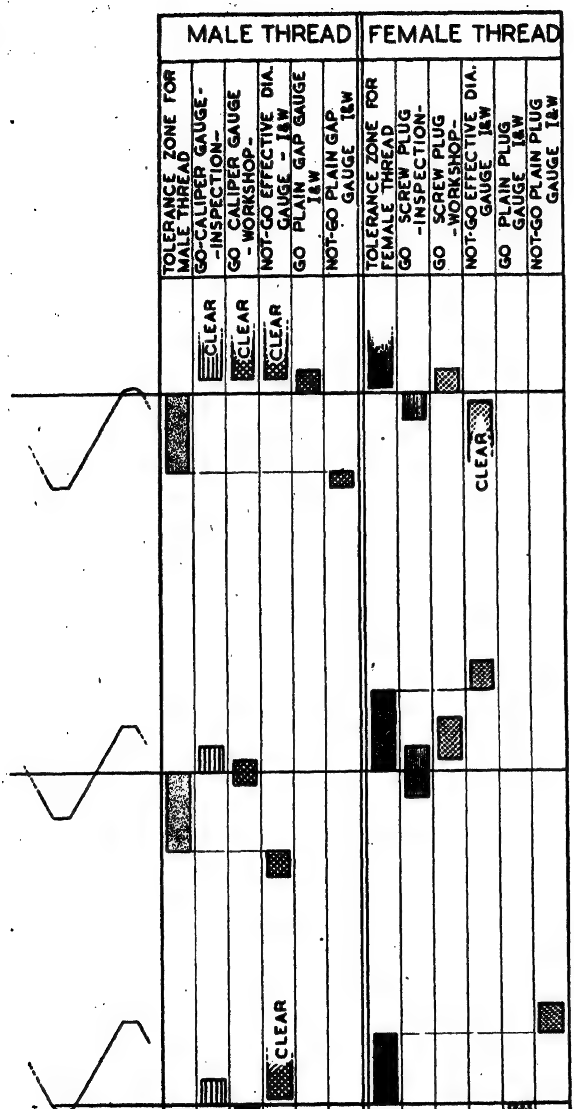


FIG. 95. THREAD TOLERANCE GAUGING DIAGRAM—METRIC AND AMERICAN 60° THREADS TO B.S.I. STANDARDS

detail in Figs. 94–98, which are self-explanatory. It is thought that British practice is the most practicable, although German practice is commendably thorough.

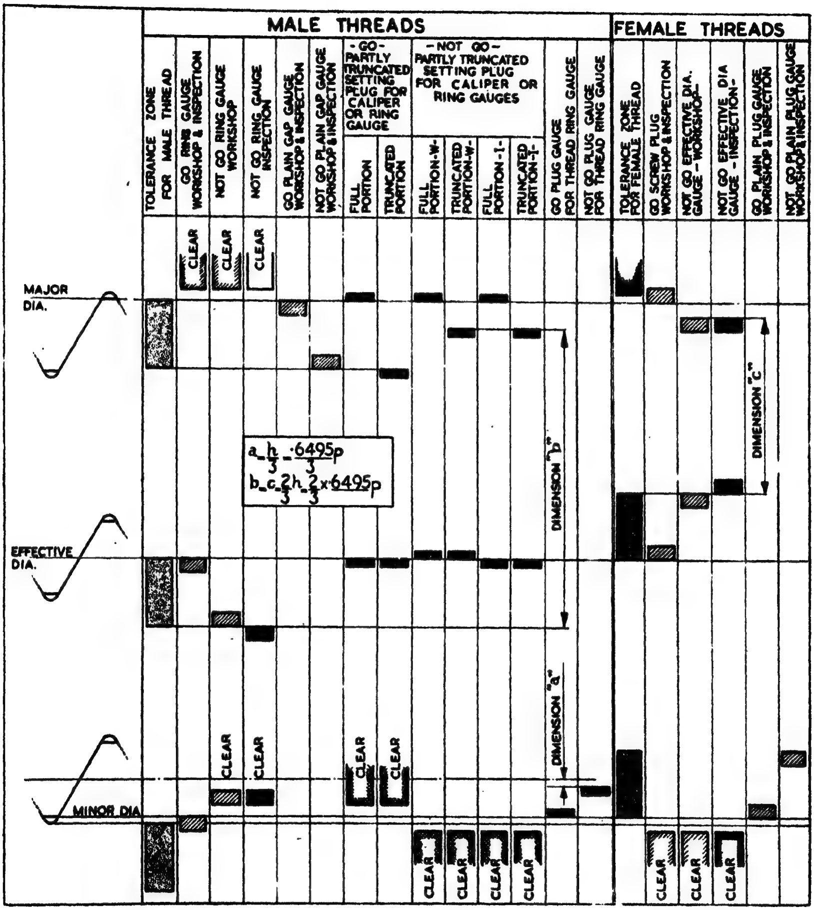


FIG. 96. THREAD TOLERANCE GAUGING DIAGRAM—AMERICAN NATIONAL THREADS

Screw Gauge Tolerances

The actual magnitudes of screw thread gauge tolerances do not directly concern the average engineer since they are small as compared with the tolerances on the threads themselves, and have little effect on the fit of the thread and its performance. For example, the effective diameter tolerance on a 1 in. B.S.F. thread is 0·0066 in.

(medium fit). The sum of the tolerance on "Go" and "Not-Go" gauges for this diameter is only 0·001 in. The effective diameter tolerance, even on the smallest B.S.F. size ($\frac{3}{16}$ in.), is 0·0033 in. for

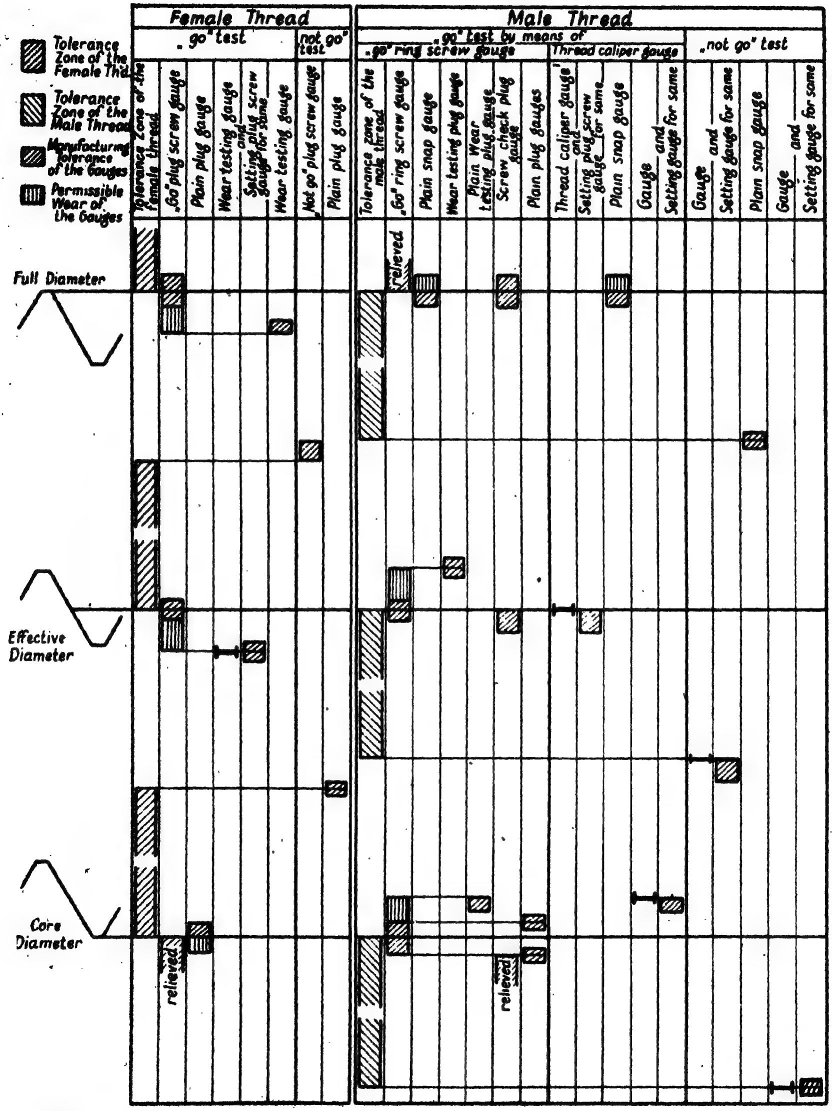


FIG. 97. THREAD TOLERANCE GAUGING DIAGRAM—GERMAN D.I.N. METRIC THREADS

medium fit or 0·0022 in. for close fit, and with work tolerances of this order, the gauge tolerance plus the error in measurement will not have any noticeable effect on the thread fit.

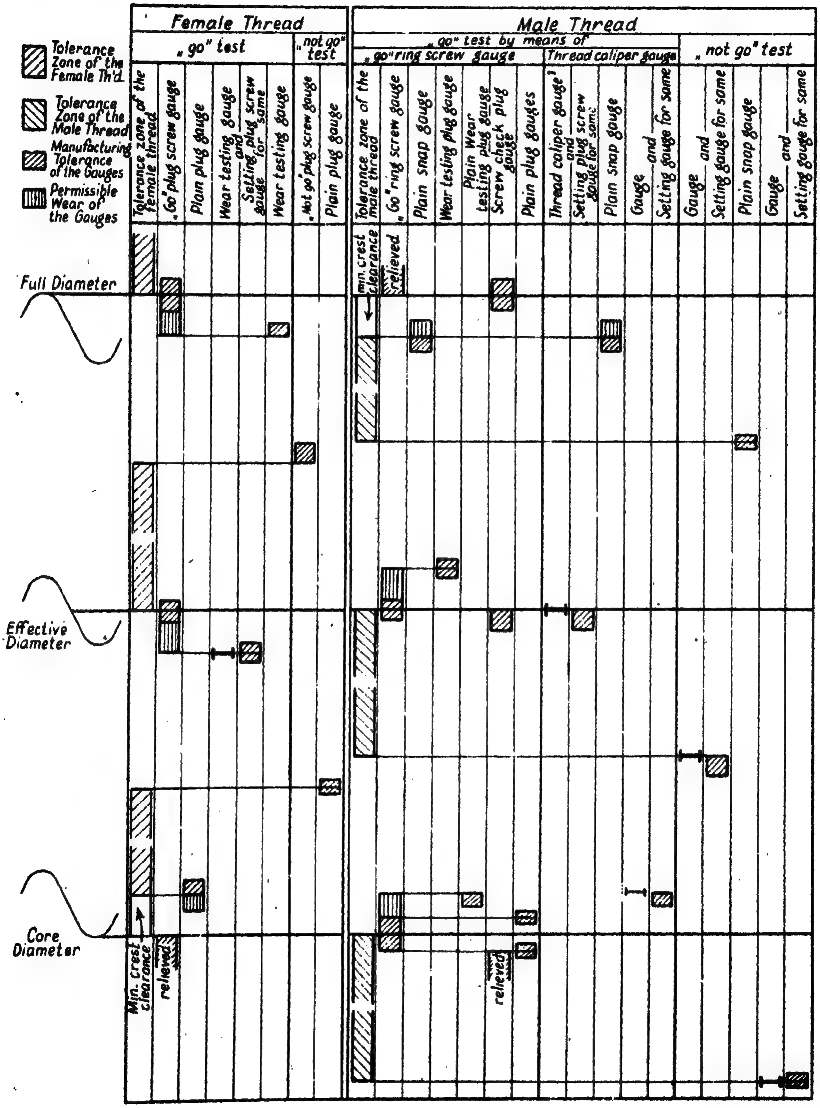


FIG. 98. THREAD TOLERANCE GAUGING DIAGRAM—GERMAN WHITWORTH THREADS

The usual disposition of screw thread gauge limits is shown in Fig. 99. It will be seen that practice differs in different countries, indicating that screw gauge limit setting has not become a refined art and that further research is necessary to arrive at the optimum system.

In the case of American limits, the work tolerance includes an allowance for the gauge limits and thus the disposition of inspection gauges inside the work tolerance does not really encroach on the theoretical intended work tolerance. The disposition of the "Not-Go" inspection outside the work tolerance, "in order to avoid

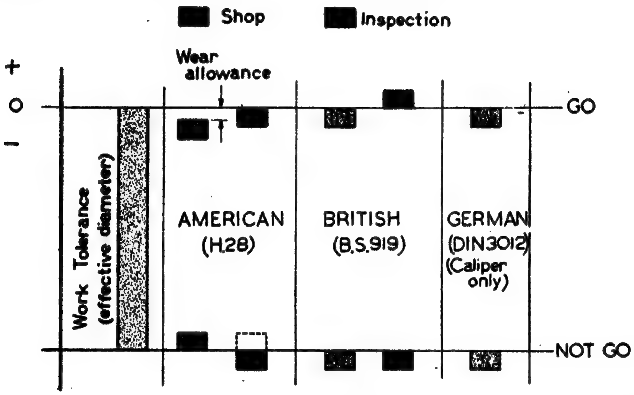


FIG. 99. DISPOSITION OF SCREW GAUGE TOLERANCES—MALE THREAD EFFECTIVE DIAMETER

needless controversy on parts close to the minimum," is permitted optionally.

The British system is similar; except that no wear allowance on male threads is made since the use of adjustable caliper gauges is recommended. For female threads, a wear allowance for the plug "Go" gauge is specified.

The German system has no separate inspection class of gauges, but lists wear limits on "Go" ring and plug gauges, though not for adjustable caliper gauges. It should be noted that, although wear allowances are not listed as such, the limits to which the gauge can wear are specified.

The precise disposition of all gauge limits for the various threads is shown in Figs. 94–98. The German Whitworth system is included for interest, but it should be remembered that this thread has crest and root clearances and hence the gauging practice must differ from B.S. practice.

CHAPTER VIII

GAUGE TOLERANCES

1. Introduction

THE more the engineer looks into gauging problems and the tolerances associated with the gauges themselves, the more cynical he must inevitably become regarding the possibilities of realizing some of the limits often quoted on drawings. This can readily be appreciated from the following simple example.

A 1 in. "B" hole to B.S. 164 has a tolerance of 0·0006 in. The standard tolerance on the gauge at both "Go" and "Not-Go" ends is 0·00005 in., allowing an "official" error of + 0·0001, + 0 to start with. With normal gauge handling errors this becomes at least 0·0002 in. Using slip blocks to set or check the gauge the tolerance on the slips themselves is introduced. A single slip to Inspection Grade blocks has a tolerance of + 7, — 3 micro-inches. Assuming a number of blocks are used wrung together, this becomes (say) + 14, — 6 micro-inches. On the other hand, Wartime Grade B blocks have limits of + 25, — 5 micro-inches, or possibly + 50, — 10 micro-inches when several blocks are added together. The error in using these slip blocks to check a plug gauge, etc., may be at least 25 micro-inches, with average labour. The "plus" error on the "Not-Go" end given above (0·0002 in.) may therefore be increased by 40–75 micro-inches, bringing the total error up to about 0·00024–0·000275 in.

The "minus" error on the "Go" end may in the same way be — 30, — 35 micro-inches. The total result is that the tolerance of 0·0006 in. may be increased by 0·0003 in. (50 per cent), or possibly even more, on a new gauge; the addition of wear on the "Go" gauge below the permitted amount is a practical possibility.

It thus becomes obvious to the designer that modern gauging practice is not up to the (at least theoretical) requirements of modern precision design. In actual practice, precision designs still need a degree of skilled assembly labour to grade and refine the drawing fits. It is also highly probable that many designs which give good performance could have their limits widened if gauging practice were improved, eliminating the errors at present existing which in themselves widen the limits.

Plain plug and caliper gauges, particularly the latter, are likely to be entirely superseded in the future for precision measurement of fine tolerances, by indicating gauges where the inspector's feel error is entirely eliminated.

2. General Considerations for all Gauges

The temperature of gauging is standardized internationally at 68° F. (20° C.).

The load under which the "Go" gauge should and the "Not-Go" gauge should not pass should always be specified. The only limit system which specifies this load properly is the I.S.A., and it must be emphasized that the error due to this type of mishandling is at least as great as the tolerance on the gauge itself, and probably as much as twice. In the example quoted above for a 1 in. "B" hole, the plug gauge may easily be forced in when 0·00005 in. oversize, and on a shaft with the same tolerance, a caliper gauge may easily be forced over 0·0001 in. undersize.

In order to eliminate controversy as to what size a limit actually refers to, it should be pointed out that it should *not* be taken as an "absolute" length as determined by some "absolute" method (e.g. light interferometry), but should be that length as set against some reference standard. By this is meant that the gap between the faces of a caliper gauge, for example, should be that arrived at by setting with a standard (e.g. slip blocks), including any error due to distortion of the gauge passing under its own (or the specified) weight and *including any oil or moisture film which may of necessity be present between the mating faces.*

Since such films are always present in practical use of setting gauges, the correct size should be defined in this manner in place of the very slightly smaller absolute size.

To be even more specific, the I.S.A. system defines the method of arriving at the proper film, the reference gauge being "glossed over with vaseline and then carefully wiped but not rubbed."

3. Published Limit Gauge Tolerance Systems

(*a*) *British Standard System* (*B.S.* 969)

This system is applicable to all inch and metric limit systems but will be applied particularly to B.S. 164 inch and metric limits and Newall inch limits. Although B.S. 164 limits date back to 1924, it is interesting to note that the gauge limits were not published until 1941. What was supposed to happen in the intervening 17 years is not clear. This is an example of the misunderstanding and ignorance prevalent on tolerance matters since it is putting the "cart before the horse." The logical method, and that adopted in the I.S.A. system, is to decide first how a limit is to be measured before determining its tolerance.

Fig. 100 shows the disposition of B.S. 969 gauge tolerances. As with other systems, the theoretical requirement that workshop

gauges shall be within inspection gauge limits has to be modified in the smaller tolerances, to avoid excessively small gauge tolerances.

It will be seen that the wear allowance is only introduced for work tolerances over 0·0035 in., in contrast with the I.S.A. system

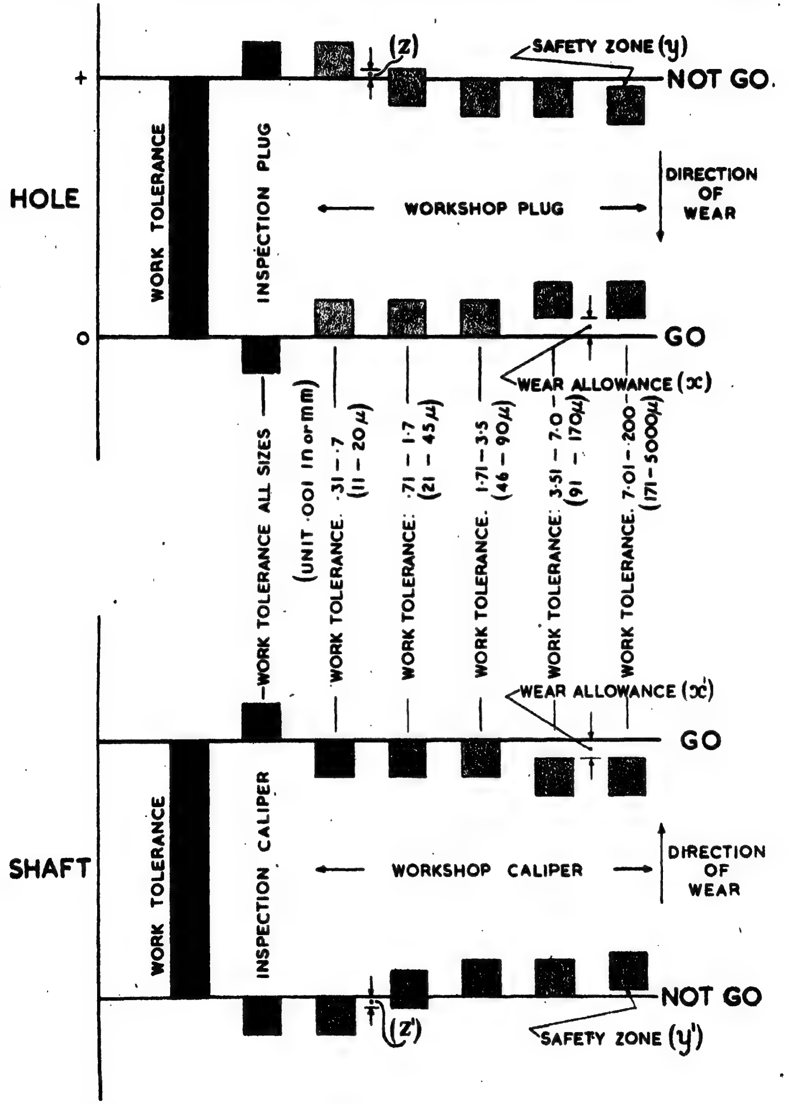


FIG. 100. DISPOSITION OF B.S. 969 LIMIT GAUGE TOLERANCES

which has a wear allowance on even the smallest work tolerance (1 micron for a work tolerance of 5 microns). Since the greatest number of tolerances are less than 0·0035 in., this is perhaps a pity. A safety zone between shop and inspection "Not-Go" gauges is introduced for work limits above 0·007 in. which are mainly on length and general gauges, not subject to such precision manufacture as small gap and plug gauges.

TABULATION OF TOLERANCES. The magnitude of workshop and inspection gauge tolerances at "Go" and "Not-Go" gauge ends is identical, but disposed differently. The disposition of these tolerances and the direction they are applied are clear from Fig. 100. Table 86 sets out all relevant data of deviations (z), wear allowance (x), and safety zones (y), as well as the tolerances themselves.

TABLE 86

B.S. 969 GAUGE TOLERANCE DATA

Inch Units (unit = 0·001 in.)					Metric Units (unit = 0·001 mm)				
Work Toler-ance	Gauge Toler-ance	z z'	x x'	y y'	Work Toler-ance	Gauge Toler-ance	z z'	x x'	y y'
0·31–0·7	0·05	—	—	—	11– 20	1·5	—	—	—
0·71–1·2	0·1	0·05	—	—	21– 30	2·5	1·5	—	—
1·21–1·7	0·15	0·05	—	—	31– 45	3·5	1·5	—	—
1·71–2·5	0·2	—	—	—	46– 65	5	—	—	—
2·51–3·5	0·3	—	—	—	66– 90	7	—	—	—
3·51–5·0	0·4	—	0·1	—	91– 130	10	—	2	—
5·1–7·0	0·5	—	0·2	—	131– 170	12	—	5	—
7·1–9·0	0·6	—	0·3	0·1	171– 230	15	—	7	2
9·1– 12	0·7	—	0·4	0·1	231– 300	17	—	10	2
12·1– 15	0·8	—	0·5	0·2	301– 400	20	—	12	5
15·1– 20	1·0	—	0·6	0·2	401– 500	25	—	15	5
20·1– 25	1·2	—	0·7	0·2	501– 600	30	—	15	5
25·1– 30	1·5	—	0·8	0·3	601– 750	40	—	20	5
30·1– 40	2·0	—	0·8	0·4	751–1000	50	—	20	10
40·1– 50	3·0	—	0·8	0·4	1001–1300	75	—	25	10
50·1– 70	4·0	—	1·0	0·5	1301–1800	100	—	30	10
70·1–100	5·0	—	1·0	0·5	1801–2500	125	—	40	15
100·1–150	6·0	—	2·0	1·0	2501–3500	150	—	50	20
150·1–200	7·0	—	3·0	1·0	3501–5000	180	—	80	25

(b) *The I.S.A. Gauge Tolerance System*

The I.S.A. Tolerance System publishes very clear and comprehensive recommendations regarding the gauges to be used and their limits, tolerances, and wear allowances. Some aspects of the system differ materially from British and American practice; for example, inspection gauges are not included, since "they are not used in

most Continental countries." The system is well thought out, however, and is intended to be revised as manufacturing technique progresses.

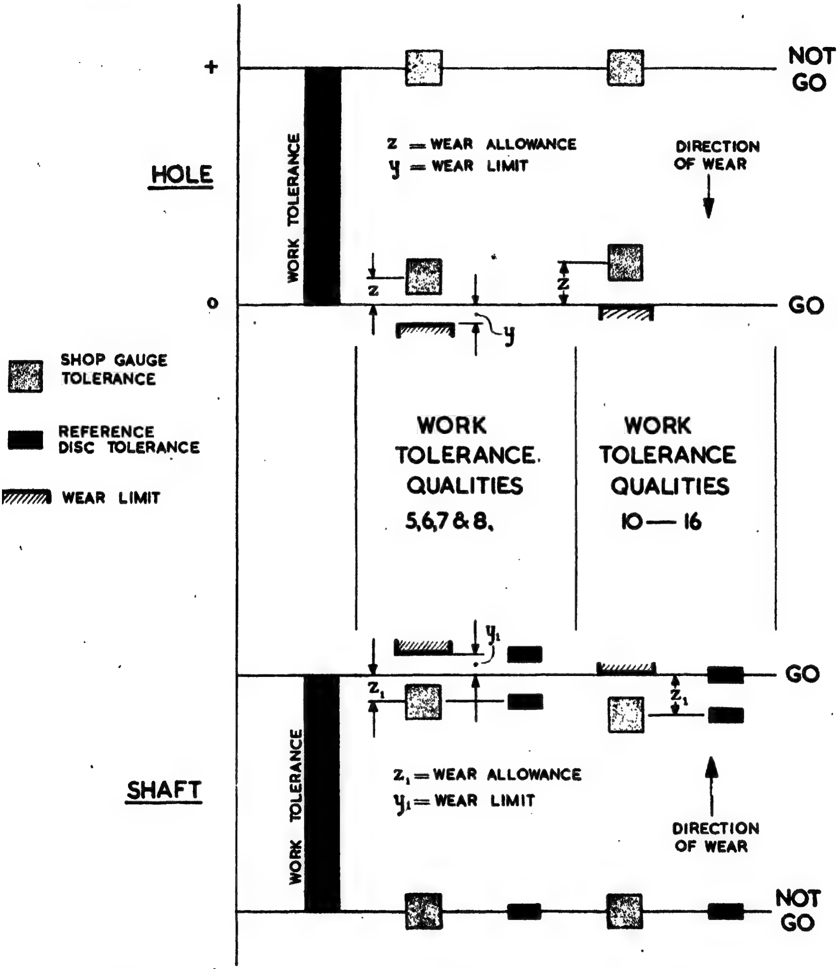


FIG. 101. DISPOSITION OF I.S.A. LIMIT GAUGE TOLERANCES (up to 180 mm)

The gauging practice recommended consists of plug and gap gauges for the "Go" side, with plug and gap gauges as well for the "Not-Go" side, except that due attention is drawn to the need for plug "Not-Go" gauges of the rod or pin type for finding the

maximum dimension of an oval or otherwise irregular hole. It is indicated that, unless otherwise stated, errors of form should be within the extreme limits quoted.

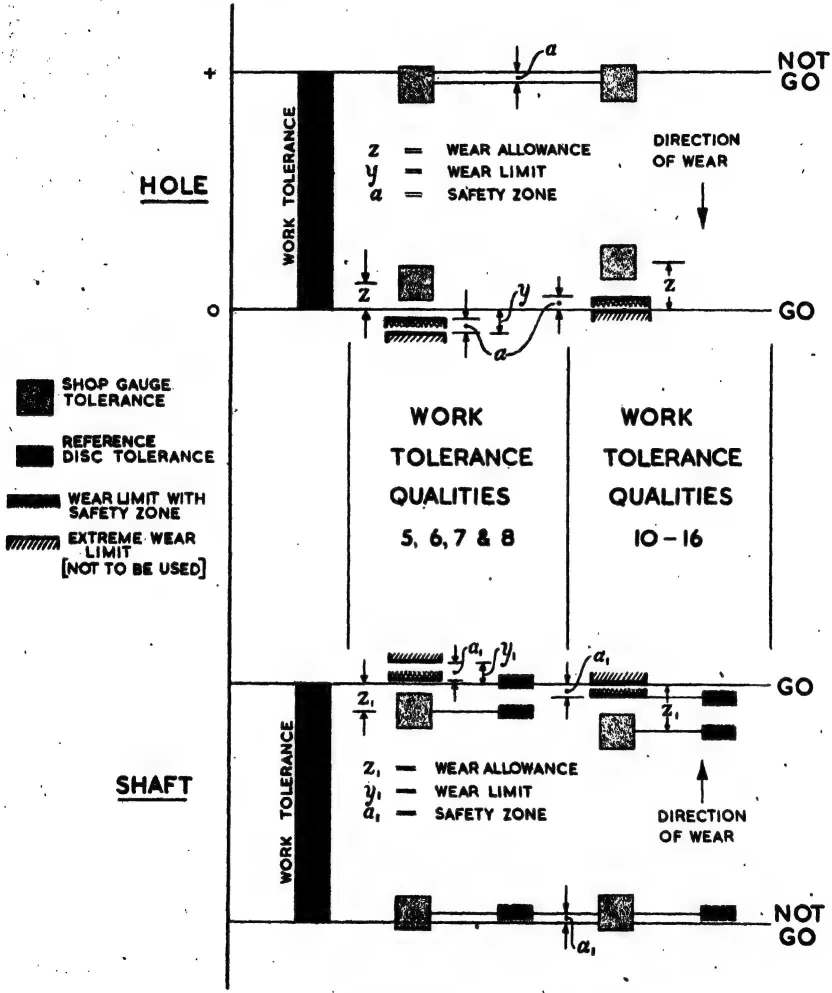


FIG. 102. DISPOSITION OF I.S.A. LIMIT GAUGE TOLERANCES (above 180 mm)

The normal gauge is the working (i.e. shop) gauge and in the case of gap gauges, reference gauges (i.e. discs) are called for, to check the "Go," "Not-Go," and wear limits. The principle of inspection is that any part passing a shop gap gauge is acceptable,

and in case of dispute the reference discs are used, the gauge to pass over the discs under its own weight. Similarly for a plug gauge, except that its size is checked by some standard means. It is said that fixed gauges are not used on the machine, although this is not altogether in accord with British and American practice, and this accounts for the use of a single set of gauges for inspection.

In order to guard against errors in measurement with fixed gauges in the large sizes (above 180 mm = 7·1 in.), a safety zone is introduced between both "Go" and "Not-Go" gauge sizes and their nominal position, thus reducing the work tolerance slightly.

DISPOSITION OF LIMITS AND TOLERANCES. Figs. 101 and 102 show diagrammatically the disposition of all gauge limits and tolerances. In these diagrams—

z = nominal wear allowance deviation of "Go" gauge, inside the work tolerance (HOLE)
z_1 = as above, but for SHAFT
y = extreme wear limit deviation of "Go" gauge outside work tolerance (HOLE)
y_1 = as above, but for SHAFT
α = safety zone (over 180 min. = 7·1 in. only)—HOLE
α_1 = as above, but for SHAFT

Table 87 gives the magnitude of the tolerances on the gauges in terms of tolerance qualities as given in Tables 19 and 20, on pages 140 and 141.

TABLE 87

MANUFACTURING TOLERANCES ON LIMIT GAUGES—I.S.A. SYSTEM

Work Tolerance	Quality IT	5	6	7	8–10	11–12	13–16	
Plug gauge . . .	IT	—	2	3	3	5	7	HOLES
Pin gauge or rod with spherical ends .	IT	—	2	2	2	4	6	
Reference disc . .	IT	1	1	1	2	2	3*	SHAFTS
Gap gauge . .	IT	2	3	3	4	5	7	

* Not considered essential.

(For details of magnitude of tolerance for particular tolerance qualities (IT's), see Tables 19 and 20.)

All tolerances are divided equally to form the limits, about the work limits involved, or the work limit suitably adjusted by the wear allowance or safety zone, or about the extreme wear limit in the case of the reference gauge for shafts up to 180 mm, "Go" end.

As will be seen in the diagrams, when the work tolerance is large enough (Qualities 10–16) the y allowance is zero, or rather is included in the z factor. It will also be seen that in most cases, work slightly outside the work limits is acceptable.

The magnitude of the allowances, z, z_1, y, y_1, α, and α_1, is indicated in Tables 88 and 89; the inch conversions have been carried out by direct conversion into suitable inch units (taking into account available slip blocks), although the diameter steps have been altered in line with the translation of I.S.A. Tolerance System given in Chapter IV. It should be emphasized that Tables 88 and 89 are comprehensible only in conjunction with the diagrams in Figs. 101 and 102, since the tables have been simplified by omitting signs, etc.

TABLE 88

SCHEDULE OF DEVIATIONS, z, z_1, y, AND y_1, UP TO 180 MM (7·1 IN.)

A. METRIC SYSTEM: Unit = 1 micron (0·001 mm)

Quality IT	5		6				7		8		9 10	11 12	13 14	15 16
Diameter	z_1	y_1	z	y	z_1	y_1	z z_1	y y_1	z z_1	y y_1	z z_1	z z_1	z z_1	z z_1
1 – 3	1	1	1	1	1·5	1·5	1·5	1·5	2	3	5	10	20	40
3·01– 6	1	1	1·5	1	2	1·5	2	1·5	3	3	6	12	24	48
6·01– 10	1	1	1·5	1	2	1·5	2	1·5	3	3	7	14	28	56
10·01– 18	1·5	1·5	2	1·5	2·5	2	2·5	2	4	4	8	16	32	64
18·01– 30	1·5	2	2	1·5	3	3	3	3	5	4	9	19	36	72
30·01– 50	2	2	2·5	2	3·5	3	3·5	3	6	5	11	22	42	80
50·01– 80	2·5	2	2·5	2	4	3	4	3	7	5	13	25	48	90
80·01–120	3	3	3	3	5	4	5	4	8	6	15	28	54	100
120·01–180	3	3	4	3	6	4	6	4	9	6	18	32	60	110

B. INCH SYSTEM: Unit = 0·001 in.

Quality	5		6				7		8		9 10	11 12	13 14	15 16
Diameter	z_1	y_1	z	y	z_1	y_1	z z_1	y y_1	z z_1	y y_1	z z_1	z z_1	z z_1	z z_1
0·04 –0·16	0·04	0·04	0·04	0·04	0·06	0·06	0·06	0·06	0·08	0·1	0·2	0·4	0·8	1·5
0·161–0·315	0·04	0·04	0·06	0·04	0·08	0·06	0·08	0·06	0·1	0·1	0·25	0·5	1·0	1·9
0·316–0·5	0·04	0·04	0·06	0·04	0·08	0·06	0·08	0·06	0·1	0·1	0·3	0·6	1·1	2·2
0·501–0·8	0·06	0·06	0·08	0·06	0·1	0·08	0·1	0·08	0·15	0·15	0·3	0·6	1·3	2·5
0·801–1·25	0·06	0·08	0·08	0·06	0·1	0·1	0·1	0·1	0·2	0·15	0·35	0·7	1·4	2·8
1·26 –2	0·08	0·08	0·1	0·08	0·15	0·1	0·15	0·1	0·25	0·2	0·4	0·9	1·6	3·1
2·01 –3·15	0·1	0·08	0·1	0·08	0·15	0·1	0·15	0·1	0·3	0·2	0·5	1·0	1·9	3·5
3·16 –5	0·1	0·1	0·1	0·1	0·2	0·15	0·2	0·15	0·3	0·25	0·6	1·1	2·1	4·0
5·01 –7·1	0·1	0·1	0·15	0·1	0·25	0·15	0·25	0·15	0·35	0·25	0·7	1·3	2·4	4·5

TABLE 89

Schedule of Deviations z, z_1, y, y_1, α and α_1, above 180 mm (7·1 in.)

A. Metric System: Unit = 1 micron (0·001 mm)

Quality IT	5			6					7			8			9		10		11		12		13		14		15		16	
Diameter	z_1	y_1	α_1	z	y	α α_1	z_1	y_1	z z_1	y y_1	α α_1	z z_1	y y_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1
180·1–250	4	3	1	5	4	2	7	5	7	6	3	12	7	4	21	4	24	7	40	10	45	15	80	25	100	45	170	70	210	110
250·1–315	5	3	1·5	6	5	3	8	6	8	7	4	14	9	6	24	6	27	9	45	15	50	20	90	35	110	55	190	90	240	140
315·1–400	6	4	2·5	7	6	4	10	6	10	8	6	16	9	7	28	7	32	11	50	15	65	30	100	45	125	70	210	110	280	180
400·1–500	7	4	3	8	7	5	11	7	11	9	7	13	11	9	32	9	37	14	55	20	70	35	110	55	145	90	240	140	320	220

B. Inch System: Unit = 0·001 in.

Quality IT	5			6					7			8			9		10		11		12		13		14		15		16	
Diameter	z_1	y_1	α_1	z	y	α α_1	z_1	y_1	z z_1	y y_1	α α_1	z z_1	y y_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1	z z_1	α α_1
7·11–10	0·15	0·1	0·05	0·2	0·15	0·1	0·3	0·2	0·3	0·25	0·1	0·5	0·3	0·15	0·8	0·15	1·0	0·3	1·5	0·4	1·8	0·6	3·1	1·0	4·0	1·8	6·7	2·8	8·3	4·5
10·01–12·5	0·2	0·1	0·05	0·25	0·2	0·1	0·3	0·25	0·3	0·3	0·15	0·6	0·35	0·25	1·0	0·25	1·1	0·35	1·8	0·6	2·0	0·8	3·5	1·4	4·5	2·2	7·5	3·5	9·4	5·5
12·51–16	0·25	0·15	0·1	0·3	0·25	0·15	0·4	0·25	0·4	0·3	0·25	0·6	0·35	0·3	1·1	0·3	1·3	0·4	2·0	0·6	2·5	1·2	4·0	1·8	5·0	2·8	8·3	4·5	11·0	7·1
16·01–20	0·3	0·15	0·1	0·3	0·3	0·2	0·4	0·3	0·4	0·35	0·3	0·7	0·4	0·35	1·3	0·35	1·5	0·6	2·2	0·8	2·8	1·4	4·5	2·2	5·7	3·5	9·4	5·5	12·6	8·7

APPENDIX 1

MICROMETER READING TESTS

THE following are the results of a series of tests carried out to determine the accuracy with which a micrometer can be used and read. A small steel plunger ground round and true to an exact size of 0·37675 in., was handed to several inspectors, skilled turners, and a few female inspectors, with the request that they measure the plunger with either or both of two 0–1 in. micrometers, one a plain one and the other a tenth-vernier type. Both instruments had friction ratchet devices. The micrometers were freshly set by the usual inspection gauge room procedure, and the setting errors can be classed as normal.

The results were as follows—

Vernier Micrometer—

Maximum reading error . .	+ 0·00025 − 0·00045
No. of readings	20
Average error (arithmetic) . .	− 0·00016

Plain Micrometer—

Maximum reading error . .	+ 0·00025 − 0·00025
No. of readings	21
Average error (arithmetic) . .	− 0·000085

General Broad Conclusions to be Drawn from these Tests

(*a*) The expected accuracy of reading can be taken as $\pm$ 0·0002 for a skilled inspector.

(*b*) The micrometer is generally read low, i.e. too much tension. The friction ratchet devices do not generally cure this, as they are only effective if handled very gently; otherwise the rotational inertia of the nut causes slight "overshooting."

(*c*) It is useless to use micrometers to measure parts whose *tolerance* is less than 0·002 in., since the error in readings represents at least 20 per cent of the tolerance.

(*d*) It is equally useless to use a micrometer to measure the upper limit of a clearance shaft if this is less than 0·001 in. clear, since the error again represents 20 per cent of the actual size.

(*e*) The vernier micrometer actually was the less accurate. This may only be true in the case of a part whose actual size is 0·00025, 0·0005, or 0·00075 in., and will thus be estimated on an ordinary micrometer to the nearest ¼-division, the tendency with the vernier being to try to read too accurately.

APPENDIX 2

ERRORS IN MEASUREMENT DUE TO SURFACE INDENTATION DURING GAUGING

THE "brinelling" effect of the pointer of an indicating gauge when measuring, both in distortion of the pointer end and in indentation of the surface being measured, has been thoroughly investigated,

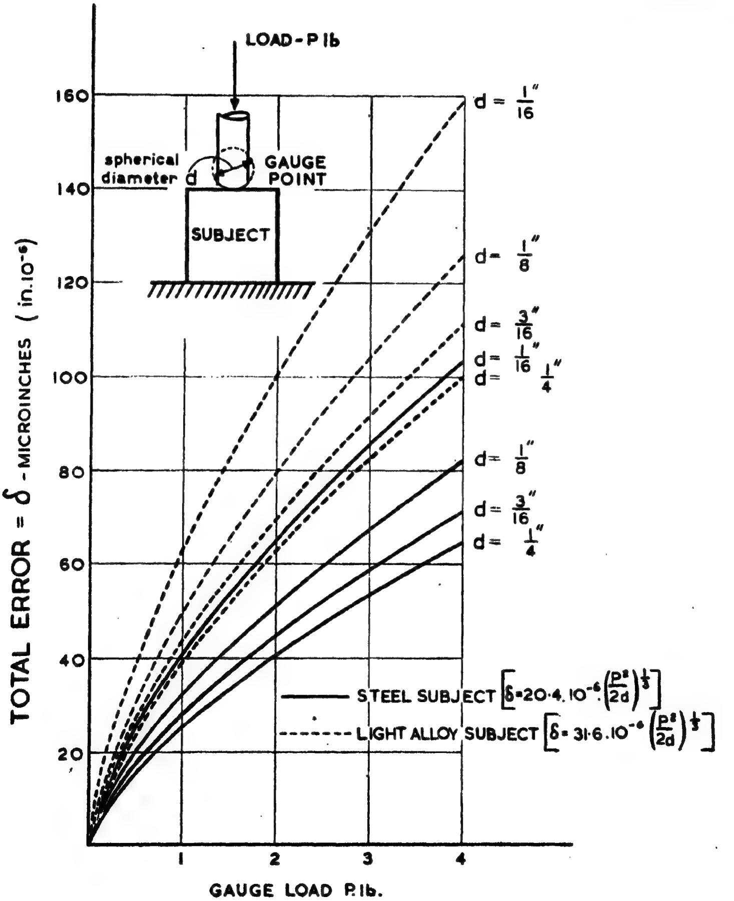


FIG. 103. CURVES OF ERROR IN GAUGING DUE TO BRINELLING

and results in not inappreciable errors in certain cases. The theoretical investigation of the problem is due to Hertz (1881), whose work forms the basis for modern ball-bearing calculations. Fig. 103 shows the error due to various gauging loads in pounds, expressed as the approach (δ) of the two bodies due to this load, for various pointer tip sphere diameters. A pointer of $\frac{1}{16}$ in. spherical diameter measuring light alloy under a load of 1 lb will read 60 micro-inches too small. Measuring a steel slip block under the same conditions causes an error of 40 micro-inches, which should be compared with a tolerance of 10 micro-inches on a standard inspection grade slip block.

The Electro-limit gauge has a diamond-ended pointer with a radius of $\frac{1}{4}$ in. and a load of 1 lb. The error measuring steel will be about 10 micro-inches, which, to say the least, is disconcerting.

The curves refer to theoretically smooth surfaces and apply accurately enough to fine gauge surfaces; for rougher surfaces, errors are likely to be increased due to local collapse of the peaks of the irregularities. Increases of as much as three times on a turned surface have been quoted.

Appendix 3

COEFFICIENTS OF LINEAR EXPANSION MODULI OF ELASTICITY

	Expansion per inch per 100° C. Temp. Rise	Young's Modulus, lb./sq. in. $\times 10^6$
Aluminium 99·4%	0·0024	10
Aluminium alloys (wrought)	0·0022	10
Aluminium bronze	0·0018	17
Brass 65/35	0·0019	14
Brass 60/40	0·0020	14
Bronze 2/10/88	0·0018	15
Copper	0·0017	18
Duralumin	0·0023	10
Gunmetal	0·0018	13
Hiduminium RR alloys	0·0022	10
Invar	0·0001	—
Iron, cast	0·0011	16
Magnesium	0·0026	6·3
Magnesium alloy (Elektron or Magnuminium)	0·0025	6·5
Monel	0·0014	25·5
Phosphor bronze 92/8	0·0017	16
Steel, cast	0·0013	29·5
Steel, structural	0·0012	29·5
Steel, alloy	0·0013	29·5
Steel, stainless 18/8	0·0017	29
Steel, stainless 20/2 (S.80)	0·0011	30
Steel, high expansion (D.T.D. 247)	0·0022	29

Appendix 4

POSITION ERRORS EXPRESSED AS ANGULAR EQUIVALENTS, FOR A RADIUS OF ONE INCH (i.e. TANGENTS OF SMALL ANGLES)

(Example: A position error of 0·0001 in. per inch = 21 seconds of a degree.)

″		′	0°	1°	″		′	0°	1°
0	0·0000000	0	0·0000000	0·0174533	31	0·0001503	31	0·0090175	0·0264708
1	0·0000048	1	0·0002909	0·0177442	32	0·0001551	32	0·0093084	0·0267617
2	0·0000097	2	0·0005818	0·0180351	33	0·0001600	33	0·0095993	0·0270526
3	0·0000145	3	0·0008727	0·0183260	34	0·0001648	34	0·0098902	0·0273435
4	0·0000194	4	0·0011636	0·0186168	35	0·0001697	35	0·0101811	0·0276344
5	0·0000242	5	0·0014544	0·0189077	36	0·0001745	36	0·0104720	0·0279253
6	0·0000291	6	0·0017453	0·0191986	37	0·0001794	37	0·0107629	0·0282162
7	0·0000339	7	0·0020362	0·0194895	38	0·0001842	38	0·0110538	0·0285070
8	0·0000388	8	0·0023271	0·0197804	39	0·0001891	39	0·0113446	0·0287979
9	0·0000436	9	0·0026180	0·0200713	40	0·0001939	40	0·0116355	0·0290888
10	0·0000485	10	0·0029089	0·0203621	41	0·0001988	41	0·0119264	0·0293797
11	0·0000533	11	0·0031998	0·0206531	42	0·0002036	42	0·0122173	0·0296706
12	0·0000582	12	0·0034907	0·0209440	43	0·0002085	43	0·0125082	0·0299615
13	0·0000630	13	0·0037815	0·0212348	44	0·0002133	44	0·0127991	0·0302524
14	0·0000679	14	0·0040724	0·0215257	45	0·0002182	45	0·0130900	0·0305433
15	0·0000727	15	0·0043633	0·0218166	46	0·0002230	46	0·0133809	0·0308342
16	0·0000776	16	0·0046542	0·0221075	47	0·0002279	47	0·0136717	0·0311250
17	0·0000824	17	0·0049451	0·0223984	48	0·0002327	48	0·0139626	0·0314159
18	0·0000873	18	0·0052360	0·0226893	49	0·0002376	49	0·0142535	0·0317068
19	0·0000921	19	0·0055269	0·0229802	50	0·0002424	50	0·0145444	0·0319977
20	0·0000970	20	0·0058178	0·0232711	51	0·0002473	51	0·0148353	0·0322886
21	0·0001018	21	0·0061087	0·0235619	52	0·0002521	52	0·0151262	0·0325795
22	0·0001067	22	0·0063995	0·0238528	53	0·0002570	53	0·0154171	0·0328704
23	0·0001115	23	0·0066904	0·0241437	54	0·0002618	54	0·0157080	0·0331613
24	0·0001164	24	0·0069813	0·0244346	55	0·0002666	55	0·0159989	0·0334521
25	0·0001212	25	0·0072722	0·0247255	56	0·0002715	56	0·0162897	0·0337430
26	0·0001261	26	0·0075631	0·0250164	57	0·0002763	57	0·0165806	0·0340339
27	0·0001309	27	0·0078540	0·0253073	58	0·0002812	58	0·0168715	0·0343248
28	0·0001357	28	0·0081449	0·0255982	59	0·0002860	59	0·0171624	0·0346157
29	0·0001406	29	0·0084358	0·0258891	60	0·0002909	60	0·0174533	0·0349066
30	0·0001454	30	0·0087266	0·0261799					

APPENDIX 5

CONVERSION TABLE FOR SMALL TOLERANCES

Microns, 0·001 mm.	To nearest Micro-inch	General Conversion, 0·001 in.	
1	39	0·04	0·05
1·5	59	0·06	
2	79	0·08	
2·5	98	0·1	0·1
3	118	0·12	
3·5	138	0·15	
4	157	0·15	
5	197	0·20	
6	236	0·25	
7	276	0·3	
8	315	0·3	
9	354	0·35	0·40
10	394	0·4	
11	433	0·4	
12	472	0·5	
13	512	0·5	
14	552	0·6	
15	590	0·6	
16	630	0·6	
17	669	0·7	
18	709	0·7	
19	748	0·75	0·7
20	787	0·8	
21	827	0·8	
22	866	0·9	
23	906	0·9	
24	945	0·9	1·0
25	984	1·0	

APPENDIX 6

TERMINOLOGY

THE following terminology in four languages is based on a supplement to the I.S.A. Bulletin 25.

English	*French*	*German*	*Italian*
1 Limits	1 Cotes limites	1 Grenzmasse	1 Dimensioni limiti
2 High limit	2 Cote maximum	2 Grösstmass	2 Dimensione (limite) massima
3 Low limit	3 Cote minimum	3 Kleinstmass	3 Dimensione (limite) minima
4 Tolerance	4 Tolérance	4 Toleranz	4 Tolleranza
5 Grade tolerance	5 Tolérance fondamentale	5 Grundtoleranz	5 Tolleranza fondamentale
6 Tolerance zone	6 Zone de tolérance (qualité)	6 Toleranzfeld	6 Campo di tolleranza
7 Manufacturing tolerance	7 Tolérance de fabrication	7 Herstellungstoleranz	7 Tolleranza di fabbricazione
8 Accuracy of manufacture	8 Précision de fabrication	8 (Herstellungsgenauigkeit)	8 Esattezza di fabbricazione
9 Tolerance unit	9 Unité de tolérance	9 Toleranzeinheit	9 Unità di tolleranza
10 Nominal size	10 Cote nominale	10 Nennmass	10 Dimensione nominale
11 Deviation	11 Écart	11 Abmass	11 Scostamento
12 Upper deviation	12 Écart supérieur	12 Oberes Abmass	12 Scostamento superiore
13 Lower deviation	13 Écart inférieur	13 Unteres Abmass	13 Scostamento inferiore
14 Actual size	14 Cote effective	14 Istmass	14 Dimensione effettiva
15 Theoretical size	15 Cote théorique	15 Sollmass	15 Dimensione teorica
16 Fit	16 Ajustement	16 Passung	16 Accoppiamento
17 Clearance	17 Jeu	17 Spiel	17 Gioco
18 Interference	18 Serrage	18 Uebermass	18 Interferenza
19 Maximum clearance	19 Jeu maximum	19 Grösstspiel	19 Gioco massimo
20 Minimum clearance	20 Jeu minimum	20 Kleinstspiel	20 Gioco minimo
21 Maximum interference	21 Serrage maximum	21 Grösstübermass	21 Interferenza massima
22 Minimum interference	22 Serrage minimum	22 Kleinstübermass	22 Interferenza minima
23 Fit	23 Cas d'ajustement	23 Sitz	23 Accoppiamento
24 Class of fit	24 Genre d'ajustement	24 Sitzart	24 Tipo di accoppiamento
25 System of fits	25 Système d'ajustements	25 Passystem	25 Sistema di accoppiamenti
26 Basic hole	26 Alésage normal	26 Einheitsbohrung	26 Foro base
27 Basic shaft	27 Arbre normal	27 Einheitswelle	27 Albero base
28 Reference line	28 Ligne zéro	28 Nullinie	28 Linea dello zero
29 Grade of fit	29 Qualité	29 Gütegrad	29 Qualità
30 Tolerance of fit	30 Tolérance d'ajustement	30 Passtoleranz	30 Tolleranza dell' accoppiamento

English	*French*	*German*	*Italian*
31 Hole	31 Alésage	31 Bohrung	31 Foro
32 Shaft	32 Arbre	32 Welle	32 Albero
33 Diameter	33 Diamètre	33 Durchmesser	33 Diametro
34 Extreme dimension of part	34 Limite d'interchangeabilité	34 Ausserstes Werkstückmass	34 Dimensione massima del pezzo
35 Clearance fits	35 Ajustements avec jeu (Ajustements mobiles[1])	35 Bewegungssitze	35 Accoppiamenti mobili
36 Clearance fits	36 Ajustements avec jeu	36 Spielsitze	36 Accoppiamenti con gioco
37 Slide fit	37 Ajustement glissant	37 Gleitsitz	37 Accoppiamento di scorrimento
38 Transition fits	38 Ajustements incertains	38 Uebergangssitze	38 Accoppiamenti incerti
39 Light push fits	39 Ajustements appuyé (Ajustement bloqué léger[1])	39 Schiebesitz	39 Accoppiamento di spinta
40 Push fit	40 Ajustement appuyé à cheval (Ajustement bloqué moyen[1])	40 Haftsitz	40 Accoppiamento bloccato leggero
41 Drive fit	41 Ajustement à cheval	41 Treibsitz	41 Accoppiamento bloccato normale
42 Press fit	42 Ajustement serré (Ajustement bloqué dur[1])	42 Festsitz	42 Accoppiamento bloccato serrato
43 Interference fit	43 Ajustement avec serrage	43 Pressitz	43 Accoppiamento bloccato alla pressa
44 Shrink fit	44 Ajustement calibre fretté	44 Schrumptsitz	44 Accoppiamento forzato a caldo
45 Gauge	45 Calibre	45 Lehre	45 Calibro
46 Limit gauges	46 Calibres à limites	46 Grenzlehren	46 Calibri differenziali
47 Internal gauge	47 Calibre d'alésage	47 Bohrungslehre	47 Calibro per fori
48 Plug gauge	48 Tampon (Calibre tampon[1])	48 Lehrdorn	48 Calibro a tampone
49 Plate gauge	49 Jauge plate	49 Flachlehrdorn	49 Calibro a tampone piatto
50 Measuring rod with spherical ends (pin gauge)	50 Broche à bouts sphériques	50 Kugelendmass	50 Calibro a barretta con estremità sferiche
51 External gauge	51 Calibre d'arbres	51 Wellenlehre	51 Calibro per alberi
52 Snap or gap gauge	52 Calibre-mâchoires	52 Rachenlehre	52 Calibro a forchetta
53 *Go* side	53 Côté *entre* (Côté *passe*[1])	53 *Gutseite*	53 Lato *passa*
54 *Not go* side	54 Côté *n'entre pas* (Côté *ne passe pas*[1])	54 *Ausschusseite*	54 Lato *non passa*
55 *Go* gauge	55 Calibre entre (Calibre *passe*[1])	55 *Gutlehre*	55 Calibro *passa*
56 *Not go* gauge	56 Calibre n'entre pas (Calibre *ne passe pas*[1])	56 *Ausschusslehre*	56 Calibro *non passa*
57 Shop gauges	57 Calibres de fabrication	57 Arbeitslehren	57 Calibri di lavorazione
58 Inspection gauges (factory acceptance gauges)	58 Calibres de révision	58 Revisionslehren (Werkstattabnahmelehren)	58 Calibri di controllo per officina

[1] Expressions used in Switzerland.

English	*French*	*German*	*Italian*
59 Acceptance gauges (purchase inspection gauges)	59 Calibres de contrôle (Calibres de réception[1])	59 Abnahmelehren (für Besteller)	59 Calibri di collaudo
60 Reference gauges	60 Rapporteurs (Calibres de référence[1])	60 Prüflehren	60 Calibri di riscontro (riscontri)
61 Manufacturing tolerance on limit gauge	61 Tolérance de fabrication des calibres	61 Herstellungstoleranz der Lehren	61 Tolleranza di fabbricazione dei calibri
62 Permissible wear	62 Usure permise	62 Zulässige Abnutzung	62 Logoramento ammesso

[1] Expressions used in Switzerland.

Appendix 7

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